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LATHE DESIGN FOR HIGH- AND LOW-SPEED STEELS

LATHE DESIGN
FOR
HIGH- AND LOW-SPEED STEELS

A TREATISE ON
THE KINEMATICAL AND DYNAMICAL PRINCIPLES GOVERNING THE CONSTRUCTION OF METAL TURNING LATHES

WITH NOTES
TO GUIDE THE PURCHASER IN THE CHOICE OF A TOOL
AND MANY EXAMPLES FROM PRACTICE

BY
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PREFACE

THIS treatise on the kinematical and dynamical principles governing the construction of metal-turning lathes is based upon the following schemes of work :—

(*a*) *Experiments upon the durability of tool-steels*, carried out at the Manchester Municipal School of Technology, under the auspices of a joint committee, and published as a Report by the Manchester Association of Engineers in 1903.

(*b*) *Researches upon the cutting forces acting upon lathe-tools*, made with a dynamometer constructed in the Manchester Municipal School of Technology, the results of which, partially published as a paper read by one of the Authors at the Chicago meeting of the Institution of Mechanical Engineers, are now given more generally in the form of curves.

(*c*) *Data and particulars collected from practice*, by the kind response of many of the best machine-tool makers to the requests of the Authors for information.

These data are all equally necessary and fundamental to the preparation of a treatise on the rational design of lathes.

Such rational design has been hitherto non-existent; the proportioning of parts and their kinematic arrangements having proceeded upon purely empirical bases.

Until the advent of high-speed steel the necessity for a theoretical treatment was unfelt. But the new conditions imposed by the general adoption of the high-heat steel were found to have rendered obsolete the long-treasured experience and accumulated data of the tool-maker. A reasoned statement of the problems involved in lathe design, and an attempt to solve them upon a basis of experimentally ascertained fact, had consequently become imperative.

The accident of the Authors having been largely responsible for the obtaining and publishing of these experimental results transformed the task of preparing a treatise upon their application to practical designing into a duty which devolved upon them, in justice, not merely to themselves, but to the Manchester School of Technology, which had borne a large share of the expense incurred, as well as to the engineering profession at large; amongst whom, although many certainly more competent for the work might have been found, none were equally conversant with the general scope and the many details of the great field of research which had been covered.

Thus the present volume forms a natural sequence to the work which has for some years been the staple product of the machine-testing laboratory of the School of Technology. It is offered as a modest example of the way in which the Technical High School of the future may strive to advance professional knowledge and lend its aid to industrial pursuits.

The substance of the book has already appeared in large part in the columns of *The Engineer*. To the Proprietors and Editor of that journal the Authors wish here to tender their thanks for the many courtesies they have received from them.

They beg further to express their sense of indebtedness to the numerous firms of machine tool makers who have aided them with much-needed information.

JOHN T. NICOLSON.
DEMPSTER SMITH.

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LATHE DESIGN FOR HIGH- AND LOW-SPEED STEEL

ERRATA

Page 58, Table XXXII., line 1, for " *Pulley* $= l\sqrt{h}$ " read " *Pulley* $7\sqrt{h}$."
,, 60, line 1, *for* " Belt drives " *read* " The belt drives."
,, 132 ,, 12, *for* " Chapter VII." *read* " Chapter VIII."
,, 145 ,, 6 from bottom, *for* " Chaps. XL.—XLIII." *read* " Chaps. XXXIX.—XLII."
,, 203 ,, 7 from bottom, *insert* " ; " *after* " $\frac{1}{220\text{B}}$ "
,, 294, Chapter heading *should read :* " SADDLES AND SLIDE-RESTS."

to give a more efficient drive than the latter; but it does not follow that it ought to be adopted in every case. The extra cost involved in its use may turn the scale of economic advantage against the weight of technical considerations which are in its favour.

In the design of a fast headstock, again, it might at first appear, in these days of high speeds and heavy cuts, to be the whole duty of the designer to obtain not only the greatest possible range of speeds of the lathe spindle, but a variation of speed within those limits as nearly as may be continuous. When, however, the question of first cost, as well as that of economical cutting, is taken into account, it will be found that there is a best ratio of successive spindle speeds for each size of lathe, depending on the complex

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LATHE DESIGN FOR HIGH- AND LOW-SPEED STEEL

CHAPTER I

INTRODUCTORY

In order to be able to design a machine upon scientific principles it is necessary to have an acquaintance with the forces which will be brought to bear upon each of its parts when it is doing its maximum and its average amount of work. The former condition governs the strength and stiffness of its various elements, whilst the latter determines their form and nature, and the materials of which they are made, from the point of view of their durability.

Strength, stiffness, and durability are, however, not the only factors which enter into the question. The conditions of manufacture which fix the cost price, the competition by makers, and the idiosyncrasies of purchasers which settle the sale price, are also very important—frequently paramount—elements in deciding what the design shall be.

It is not sufficient, therefore, to consider the design of a machine tool in its technical and scientific aspect only; the economic or commercial point of view must also be taken into account.

It is known, for instance, that a narrow belt running at a high speed is a more efficient transmitter of power than a broad one at a low speed. This high speed may be obtained by using either large pulleys at a low speed of revolution, or small pulleys rotating rapidly. The former condition is known to give a more efficient drive than the latter; but it does not follow that it ought to be adopted in every case. The extra cost involved in its use may turn the scale of economic advantage against the weight of technical considerations which are in its favour.

In the design of a fast headstock, again, it might at first appear, in these days of high speeds and heavy cuts, to be the whole duty of the designer to obtain not only the greatest possible range of speeds of the lathe spindle, but a variation of speed within those limits as nearly as may be continuous. When, however, the question of first cost, as well as that of economical cutting, is taken into account, it will be found that there is a best ratio of successive spindle speeds for each size of lathe, depending on the complex conditions which obtain in practice; and the problem is again of the nature of a commercial compromise.

To confine our attention to the technical aspect only of such problems would be, therefore, to condemn in advance the solutions we should obtain; and it is to the constant recurrence of such incomplete treatment of engineering questions, indeed, that we must impute the disrespect with which practical men frequently regard conclusions deduced from "theory." In the last sentence of his preliminary dissertation "On the Harmony of Theory and Practice in Mechanics," Rankine says, "The engineer or the mechanic who plans and works with understanding of the natural laws that regulate the result of his operations rises to the dignity of a sage." This is undoubtedly true; but it requires to be emphasized that the phrase "natural laws" has reference quite as much to economic conditions as to considerations merely physical.

In the sequel, wherein we propose to discuss questions of the design and proportions of an article so commonly manufactured as a machine tool, it will, therefore, be the merest axiom of common sense to submit all technically obtained results to the crucial test of their fitness to survive under the normal and well-ascertained conditions of successful commercial production.

Until quite recently machine tools have been produced in a very stereotyped way, the designs having evolved themselves by the processes of trial and error, rule of thumb, and the survival of the fittest. If long and costly, such a process of unintelligent experimenting very frequently leads to a result which it would be difficult to excel. That man would indeed be worthy of the name of sage who should, from the recesses of his brain, excogitate a machine of higher efficiency, greater suitability, and smaller first cost, than any of those in actual use which have been the object of the constant thought and experience of practical men for centuries.

It is only when new conditions arise, such as the passing of the horse, the advent of high-speed steel, or the discarding of reciprocating in favour of rotary prime movers, that an appeal is made to theory or experiment for some assistance in bridging the chasm between past experience and present necessity.

Such a state of things now obtains with regard to machine tools. The rapid development and universal adoption of the new high-speed steel has necessitated the abandonment of the old standard rules of lathe construction and design, which are no longer found equal to the demands of the new material. The volume of results of experience so far compiled by single firms is too small to act as a universal guide in projecting new designs. It therefore appears to be opportune to offer a new series of rules of design based upon a long series of experiments on cutting with the new steels, and checked by numerical data collected from the most recent practice of a number of eminent firms of machine tool makers.

The deductions of a theory sound both technically and commercially cannot but be of value to those engaged in the design and manufacture of a class of machines which is at the present time in a state of transition owing to the new departure in tool steel production. It is hoped also that, if the purchaser can be reached, he may be to some extent educated to a higher sense of responsibility in regard to altering standard lines of design, unless for proper and sufficient reason; and, on the other hand, he will be told what he ought to expect in a lathe of a given size in the way of cutting power, spindle speeds, gear ratios, belt or motor drive, and the numerous details of design on which it is possible to give a verdict for or against.

CHAPTER II

HISTORICAL

It seems but yesterday that visitors to the Paris Exhibition were astonished by the sight of an American lathe cutting great spiral shavings of steel which came off at a blue heat, whilst the point of the tool itself was visibly red-hot in the daylight. Yet there are now many firms in Britain which produce tool steels capable of much greater feats of endurance and efficiency.

If great credit is due to Messrs. Taylor and White for the great advance they made in perfecting Mushet's discovery of self-hardening steel, it must also be admitted that the British steel makers have spared no expense of time or talent in following the matter up, and obtaining a record for quality and uniformity and ease of manipulation of their steels which is now nowhere surpassed, if, indeed, it be equalled.

There are two distinct kinds of tool steel, called respectively water-hardened, or carbon steel; and air, or self-hardening steel, which was discovered by R. Mushet in 1860. Mushet found that by adding manganese and tungsten to the steel, it maintained its cutting edge at much higher temperatures of the tool point; and that, consequently, much higher speeds of cutting than had previously been attained were possible. This steel came, therefore, into very general use, and completely replaced carbon steel for roughing out; but the great possibilities latent in tungsten steel were not revealed until Messrs. Taylor and White undertook a long series of experiments to investigate the phenomena of the process of hardening it, at the works of the Bethlehem Steel Company, Pa.

Up to this time all air-hardened steel had been invariably heated only up to a cherry-red temperature—about 1550° F.—and users were cautioned against heating beyond this before allowing the tool to cool gradually or in a blast of air. What Messrs. Taylor and White discovered was that when air-hardening steels are made with certain constituents in ascertained proportions they are greatly improved in their cutting efficiency by being heated to much higher temperatures than had ever before been tried in the process of hardening. They found that, although the steel appeared to be injured by heating to any temperature between 1550° and 1725° F., above the latter temperature and up to about 2000° F.—when the steel softens or crumbles when touched with a rod—the efficiency was greatly increased. If the greatest safe cutting speed for steel heated to cherry-red was 30 feet per minute, then, when recourse was had to heating up to the "burning" point, the allowable speed would be increased to about 80 or 90 feet per minute. This discovery of Messrs. Taylor and White has proved to be the beginning of a new epoch in machine practice. It has brought about the running of machines at much higher speeds than have hitherto been thought possible; and by demanding closer speed regulation with greater range and a greater attention to the conditions governing power, strength, and rigidity, it has created a new era in machine-tool design.

As the result of their experiments, the Bethlehem Steel Company were able to perform great feats in the way of high temperature and high-speed cutting in 1900; but their results were obtained on very soft material, and doubts were at first entertained as to the practical value of the discovery in general engineering work.

Numerous tests were, in consequence, set on foot in workshops all over the world, but the opinions formed as the result of such sporadic trials were so conflicting that the Berlin section of the Verein Deutscher Ingenieure decided to take the matter up; and in the spring of 1901 they set out to ascertain the value and applicability of the new steel in practical work. The results of their extensive series of experiments were published in the *Zeitschrift* for September 27, 1901.

BERLIN EXPERIMENTS

The object of the trials was to determine, keeping in view the durability or life of the tools: (1) What maximum surface could be machined in unit of time with a given cut of $\frac{3}{16}$ in., the traverse and speed being at the option of the experimenters. (2) What weight of cuttings could be removed per

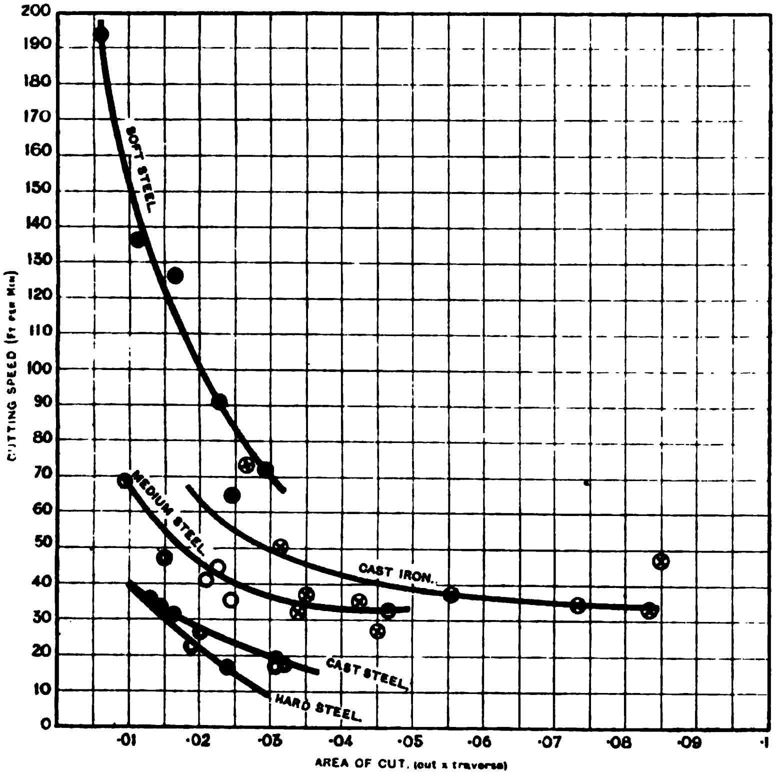

FIG. 1.—BERLIN TOOL STEEL TESTS.

unit of time with the greatest possible depth of cut, the traverse feed and speed being again left open.

The materials operated upon were grey cast iron, cast steel, and forged or rolled Siemens-Martin steel of various ultimate tenacities. The tool steels were those supplied by the three firms, Bergische Stahl Industrie, of Remschied; Gebrüder Böhler and Co., of Vienna and Berlin; and Poldi-Hütte, of Vienna and Berlin. About two hundred and fifty trials, of duration up to two hours each, were made, and a selection of the best results is represented in Fig. 1, where fair curves are drawn through the spots obtained for each material.

These curves give the relation between the cutting speeds and the areas of cuts for which the tools endured at least one hour—with one or two exceptions—without requiring regrinding. The curve marked "soft steel" is for cuts taken upon Siemens-Martin steel of 26 tons tenacity; whilst the "medium steel" and "hard steel" curves were for the same material, but with ultimate strengths of 40¼ and 49 tons respectively.

The analyses and other properties of the various materials operated upon are summarized in the annexed Table I.; and this table contains the same figures for the materials used also in the Manchester experiments to be referred to below:—

TABLE I

Chemical Composition and Strength Properties of (A) = Cast Iron used in Berlin and Manchester

	Berlin.	Manchester.		
Carbon (total)	3·91	—	—	—
„ (combined) . .	0·45	0·459	0·585	1·150
Graphite	3·46	2·603	2·720	1·875
Silicon	2·05	3·010	1·703	1·789
Manganese	1·00	1·180	0·588	0·348
Sulphur	0·10	0·031	0·061	0·1614
Phosphorus	0·10	0·773	0·526	0·782
Crushing strength . . .	—	26·9 tons per sq. in.	44·0 tons per sq. in.	43·5 tons per sq. in.
Name		"Soft"	"Medium"	"Hard"

(B) = Steel used in Berlin and Manchester.

	Berlin. Siemens-Martin.			Manchester. Whitworth fluid-pressed.		
Carbon . .	0·30	0·54	0·63	0·198	0·275	0·514
Silicon . .	0·05	0·21	0·20	0·055	0·086	0·111
Manganese .	0·58	0·93	1·22	0·605	0·650	0·792
Sulphur . .	0·05	0·025	0·05	0·026	0·037	0·033
Phosphorus	0·07	0·05	0·05	0·035	0·043	0·037
Tenacity . .	26 to 32 tons per sq. in.	40¼ tons per sq. in.	49 tons per sq. in.	26·3 tons per sq. in.	28·9 tons per sq. in.	46·7 tons per sq. in.
Yield point .	—	—	—	12·8 tons per sq. in.	9·8 tons per sq. in.	19·7 tons per sq. in.
Name . . .	"Soft"	"Medium"	"Hard"	"Soft"	"Medium"	"Hard"

MANCHESTER EXPERIMENTS.

These experiments were begun at the Manchester Municipal School of Technology in October, 1902, and extended over ten months. The expense incurred was jointly borne by the Manchester Association of Engineers, by the School of Technology, and by the firm of Sir W. G. Armstrong, Whitworth, and Co. The report upon them was presented at a meeting of the association on October 24, 1903, and may be obtained from the secretary. The working members of the committee responsible for the carrying out of the experiments were Mr. Daniel Adamson, to whose energy and ability the success of the experiments were largely due, Mr. H. N. Bickerton, Mr. E. G. Constantine, Mr. George Daniels, and one of the Authors.

The questions which the committee set themselves to answer in these trials were—

(1) What maximum cutting speeds can be obtained with the new steels when taking light or finishing cuts upon hard, medium, and soft steel, and upon hard, medium, and soft cast iron?

(2) What maximum area of surface can be machined in a given time when taking a $\frac{3}{16}$-in. cut with the new steels upon the six given materials?

(3) What greatest weight of cuttings can be removed in a given time from these three grades of steel and of cast iron?

(4) What forces are operative upon the tool in making these cuts, and according to what law do these forces vary with the speed of cutting and the area and shape of the cut?

(5) Can the new steels be forged and tempered by an ordinary smith, and yet be relied upon to give results as to cutting speed and durability similar to those obtained when they are delivered ready ground by the makers?

The answers to the first four of these questions are concisely given in Table II., which has been abstracted from the results of the report. The table is self-explanatory, and it need only be added that "non-failing tools" are those which were capable of further cutting at the expiration of twenty minutes; and tools which failed were those whose withdrawal was necessary before the expiration of that time. In the last two columns are given the values of the cutting forces and the cutting stresses—or forces divided by areas—for each variety of steel or cast iron, irrespective of the size or shape of the cut taken; the assumption that the cutting force is proportional to the area of the cut being, as a first approximation, assumed to hold.

Figs. 2 and 3 give these results as originally published in the report, and the curves plotted in them will be found to be generally in close agreement with the numerical results of Table II.

The speeds plotted on a base of area of cut are those at which the tool must be run for the corresponding cuts, if it is to last equally long in each case. The formulæ found most closely to represent the relation between speed and area were of the form (*vide* Report)—

$$v = \frac{K}{a + L} + M$$

v being the cutting speed in feet per minute, a the area of the cut in square

TABLE II

Manchester Experiments

Intended cut and traverse for all tools, inches.		Actual area of cut, square inches.		Cutting speed, feet per minute.		Weight removed, lb. per minute.		Force and stress.	
		Non-failing tools of greatest speed.	Average of all non-failing tools.	Greatest speed of any non-failing tool.	Average speed of all non-failing tools.	Greatest weight removed by any non-failing tool.	Average weight removed by all non-failing tools.	Average cutting force of all non-failing tools.	Average cutting stress of all non-failing tools, tons per square inch.
Soft Steel (Whitworth Fluid-pressed).									
$\frac{1}{16}$	$\frac{1}{16}$	0·0032	0·0035	149·2	125·9	1·72	1·455	921	117
$\frac{3}{16}$	$\frac{1}{16}$	0·0106	0·0104	111·0	95·8	4·14	3·42	2,590	111
$\frac{3}{16}$	$\frac{1}{8}$	0·0215	0·0210	74·0	62·8	5·28	4·38	5,454	115
$\frac{3}{8}$	$\frac{1}{8}$	0·0434	0·0422	50·7	44·5	7·5	6·26	11,740	124
Medium Steel (Whitworth Fluid-pressed).									
$\frac{1}{16}$	$\frac{1}{16}$	0·0039	0·0036	105·2	101·7	1·5	1·295	925	115
$\frac{3}{16}$	$\frac{1}{16}$	0·0114	0·0115	80·0	71·8	3·17	2·8	2,743	106½
$\frac{3}{16}$	$\frac{1}{8}$	0·0227	0·022	51·4	49·9	3·88	3·68	5,742	116
$\frac{3}{8}$	$\frac{1}{8}$	0·0425	0·0428	38·6	37·8	5·60	5·47	10,437	109
Hard Steel (Whitworth Fluid-pressed).									
$\frac{1}{16}$	$\frac{1}{16}$	0·0040	0·0037	52·5	51·3	0·73	0·66	1,274	154
$\frac{3}{16}$	$\frac{1}{16}$	0·0124	0·012	41·2	40·6	1·71	1·64	3,658	136
$\frac{3}{16}$	$\frac{1}{8}$	0·0219	0·0219	30·8	30·8	2·3	2·3	7,960	162
$\frac{3}{8}$	$\frac{1}{8}$	0·0452	0·0438	20·2	19·6	3·1	2·91	15,220	155
Soft Cast Iron.									
$\frac{1}{16}$	$\frac{1}{16}$	0·0039	0·00397	109·0	105·25	1·3	1·27	489	55
$\frac{3}{16}$	$\frac{1}{16}$	0·0124	0·0123	99·5	85·26	3·6	3·1	1,051	38
$\frac{3}{16}$	$\frac{1}{8}$	0·0214	0·0210	66·2	65·1	4·38	3·97	2,555	54
$\frac{3}{8}$	$\frac{1}{8}$	0·0459	0·0456	55·5	50·75	7·46	7·02	4,403	43
Medium Cast Iron.									
$\frac{1}{16}$	$\frac{1}{16}$	0·0033	0·0035	59·7	54·1	0·69	0·65	818	104
$\frac{3}{16}$	$\frac{1}{16}$	0·0115	0·0107	49·0	44·6	1·73	1·51	2,045	85
$\frac{3}{16}$	$\frac{1}{8}$	0·0227	0·0201	33·1	32·5	2·23	2·07	3,630	80½
$\frac{3}{8}$	$\frac{1}{8}$	0·0446	0·0443	24·3	23·2	3·32	3·13	6,085	61
Hard Cast Iron.									
$\frac{1}{16}$	$\frac{1}{16}$	0·00375	0·0037	38·5	37·6	0·48	0·45	953	115
$\frac{3}{16}$	$\frac{1}{16}$	0·01173	0·011	31·9	29·7	1·18	1·04	2,009	81
$\frac{3}{16}$	$\frac{1}{8}$	0·02112	0·0204	24·7	22·74	1·62	1·39	3,772	83
$\frac{3}{8}$	$\frac{1}{8}$	0·0459	0·0432	22·0	19·0	3·2	2·56	5,443	56

inches, and K, L, and M constants, whose physical significance need not be further discussed at this point, but whose numerical values for the various materials operated upon are as follows:—

TABLE III

Constant.	Fluid pressed steel.			Cast-iron bars.		
	Soft.	Medium.	Hard.	Soft.	Medium.	Hard.
K	1·95	1·85	1·03	3·10	1·65	1·30
L	0·011	0·016	0·016	0·025	0·030	0·035
M	15	6	4	8	7	5·5

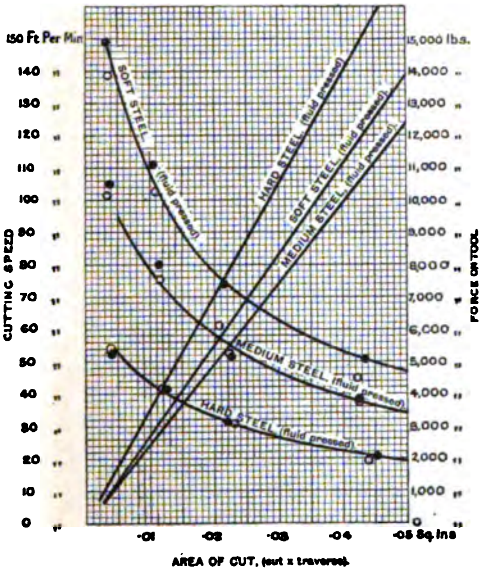

Fig. 2.—Manchester Tests—Steel.

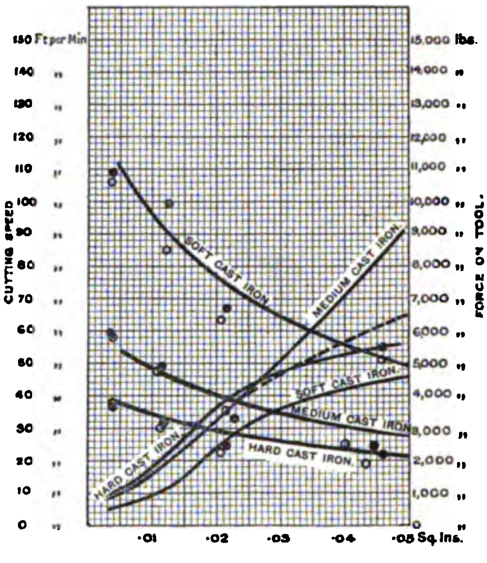

Fig. 3.—Manchester Tests—Cast Iron.

In Fig. 4 the speed-on-area results from both the Berlin and Manchester trials have been plotted together—open dots = Berlin, full dots = Manchester. The thick black curve depicts the relation between cutting speed and area of cut given by the formula—

$$v = \frac{1}{a} + 15 \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

which will be very frequently used in the sequel (*vide* Chapter VI.). Here v = cutting speed in feet per minute, and a = area of cut in square inches.

This is the formula for the speed at which a given cut is assumed to be taken under ordinary workshop conditions, and at which the heaviest cut for which a given lathe is designed must be taken, when under test.

It is seen to be well within the highest values obtained in the Manchester experiments, but passes close to those obtained in Berlin for the lightest cuts. This is justified by the consideration that the Berlin trials give speeds for which the tools lasted over one hour, whilst the Manchester results are for tools intended to last only twenty minutes. The so-called "soft steel" used in the Manchester trials was very tough, and it is therefore possible that the constant—in Formula (1)—may be taken as high as twenty-five for ordinary "machinery steel," without an undue amount of destruction of the tool being encountered in ordinary practice.

The round-dot curve traced on Fig. 4 shows the relation which would have obtained between speed and area had it been experimentally found that the speed of cutting could increase as fast as the area of the cut diminished, *i.e.* that the same weight could be machined off per minute for a light as for a heavy cut. We see at a glance that when the area is reduced from 0·1, with a speed of 33 ft. per minute, to 0·05 square inch, *e.g.* the speed, instead

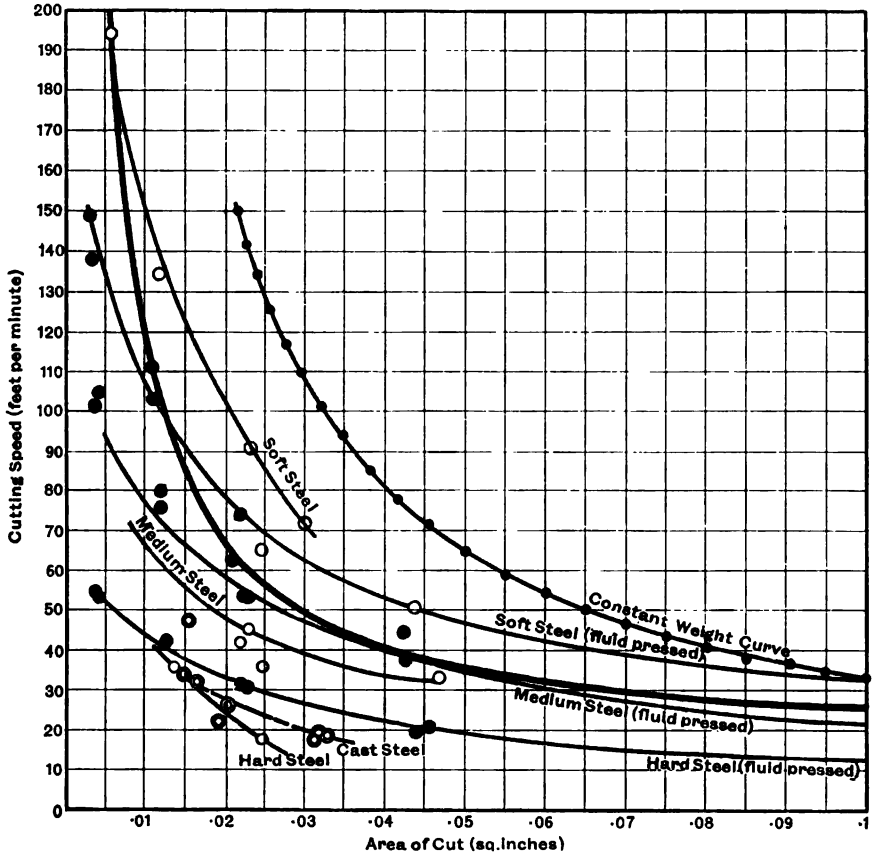

Fig. 4.—Tool Steel Tests—Berlin and Manchester.

of being increased 100 per cent., can only be raised by 42 per cent., and that if reduced to 0·025 square inch the speed can only be about doubled, instead of being increased fourfold. The weights removed will, therefore, be $\frac{3}{4}$ and $\frac{1}{2}$ respectively, in the two latter cases, of what could be removed when the cut was 0·1 and the speed 33. For light cuts, such as $\frac{1}{16}$ in. by $\frac{1}{16}$ in., of area 0·005, weight can be machined off which only amounts to 20 per cent. of what can be removed with a cut of 1 in. by $\frac{1}{10}$ in. traverse.

In Fig. 5 are shown curves which fairly represent the weights removed per minute in the committee's experiments and in subsequent trials. It is inserted here for convenient reference.

The full lines are for steel; the dotted are for cast-iron. The spots give the actual results obtained, being the greatest weights removed by any non-failing tool on each of the four series of trials made with the six samples operated upon. As ordinates we have pounds removed per minute, as abscissæ area of cut—or depth of cut by width of traverse. Since W = MaV, where M is the density of the material, a the area of cut, V the speed, and W the pounds removed per unit of time, we have $\frac{W}{a} = MV = \tan a$, say. This means that if a sloping line be drawn through the origin or co-ordinates of the diagram of weights and cuts, making with the axis of areas an angle a, whose tangent is MV, this sloping line will join points, giving the weights

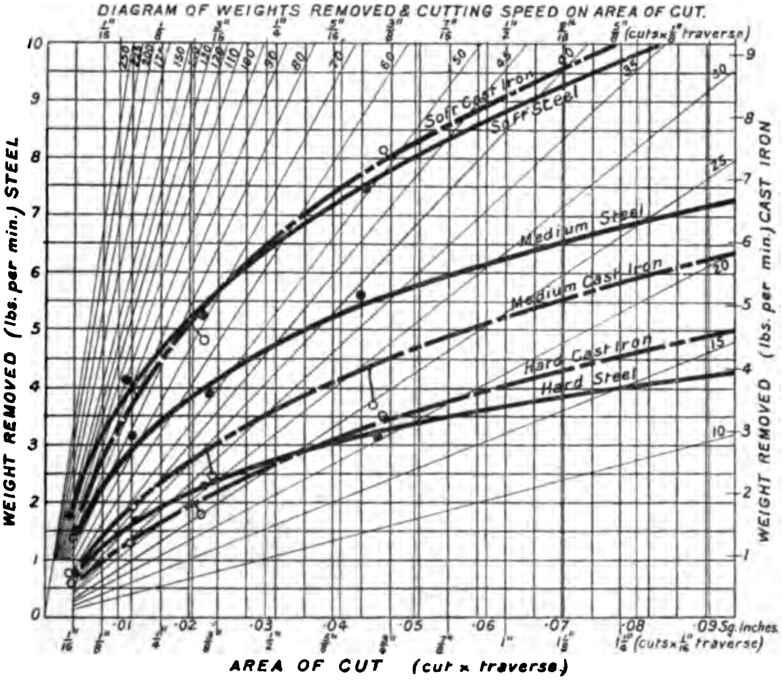

FIG. 5.—WEIGHT OF METAL REMOVED.

removed for each cut at the given speed V—if this be a possible speed. Such sloping lines will, in fact, be (contour) lines of constant speed, and are a useful addition to the diagram of weights removed. Thus—*e.g.* if we wish to know at what speed 5 lb. per minute can be removed, we can tell at a glance by running across the diagram upon the 5 lb. line that the speed would have to be 200 ft. per minute for ⅛ in. by 1/16 in. cut (0·00391 area), 96 ft. per minute for ⅛ in. by ⅛ in. cut (0·0156 area), and 33 ft. per minute for a ⅜ in. by ⅛ in. cut (0·047 area).

The first speed is impossible on materials operated on, the second is problematic, and the third could only be taken upon the milder varieties of steel. Or, conversely, we can tell the weights which will be removed if we run at a given speed on a given material; also the depth of cut and width of traverse we may safely employ at that speed.

CHAPTER III

CUTTING FORCES—METHODS OF EXPERIMENTING

In the trials of the Manchester committee, which we described in our second chapter, there appeared an entire lack of uniformity in the shapes and angles of the tools submitted by the eight competing firms of steel makers. There was also no obvious connection between the shapes and angles of the tools and the cutting forces exerted by these tools as deduced in the report from the electrical power measurements made by the committee. Neither did the shape or angle supply a clue to the causes of success and failure in the various durability trials with different tools.

The type of lathe for heavy and rapid cutting, which was soon found to be generally necessary owing to the universal adoption in practice of the improved steel, called, on the other hand, for a reconsideration of the whole question of lathe design from the point of view of the forces exerted by the tool. Approximately correct values for the vertical cutting force at the heavier cuts had been obtained by the Manchester committee; but the magnitude of the forces required for the traversing and surfacing movements of the slide rests were quite unknown.

It was also very desirable to have a more definite knowledge of the law of variation of these forces with the shape and size of the cut, and with the shape and cutting angle of the point of the tool.

A special lathe-tool dynamometer was therefore designed and constructed at the Manchester Municipal School of Technology for the more exact measurement of the quantities in question by a direct method, instead of as the relatively small difference of two measurements of electrical horse-power.

Thanks are due to the authorities of that school for defraying the not inconsiderable expense incurred for power, light, and mechanical assistance, and to Messrs. Sir W. G. Armstrong, Whitworth and Co., for the lathe and materials used in the prosecution of a somewhat extensive research.

The Lathe Dynamometer.—This has been already fully described in a paper read by one of the authors at the Chicago (1904) meeting of the Institution of Mechanical Engineers [*Proc.* 1905], and a short description only need here be given, with three views redrawn to a larger scale. After considerable experience it was found possible to measure, without undue chattering, cutting forces up to magnitudes of about ten tons, with the tool so suspended as to be capable of freely moving in all three co-ordinate directions, except as constrained by the measuring apparatus. From the views shown on this page, it will be seen that the tool can turn about an axis (parallel to that of the work) formed by the points of two screws Q which pass through a pair of sliding blocks W, fitting in slots in the casting O. This casting is recessed to provide for the gimbal tool-holder P, which pivots on those pointed screws Q. Two similar screws R, see Fig. 7, pass through the gimbal into

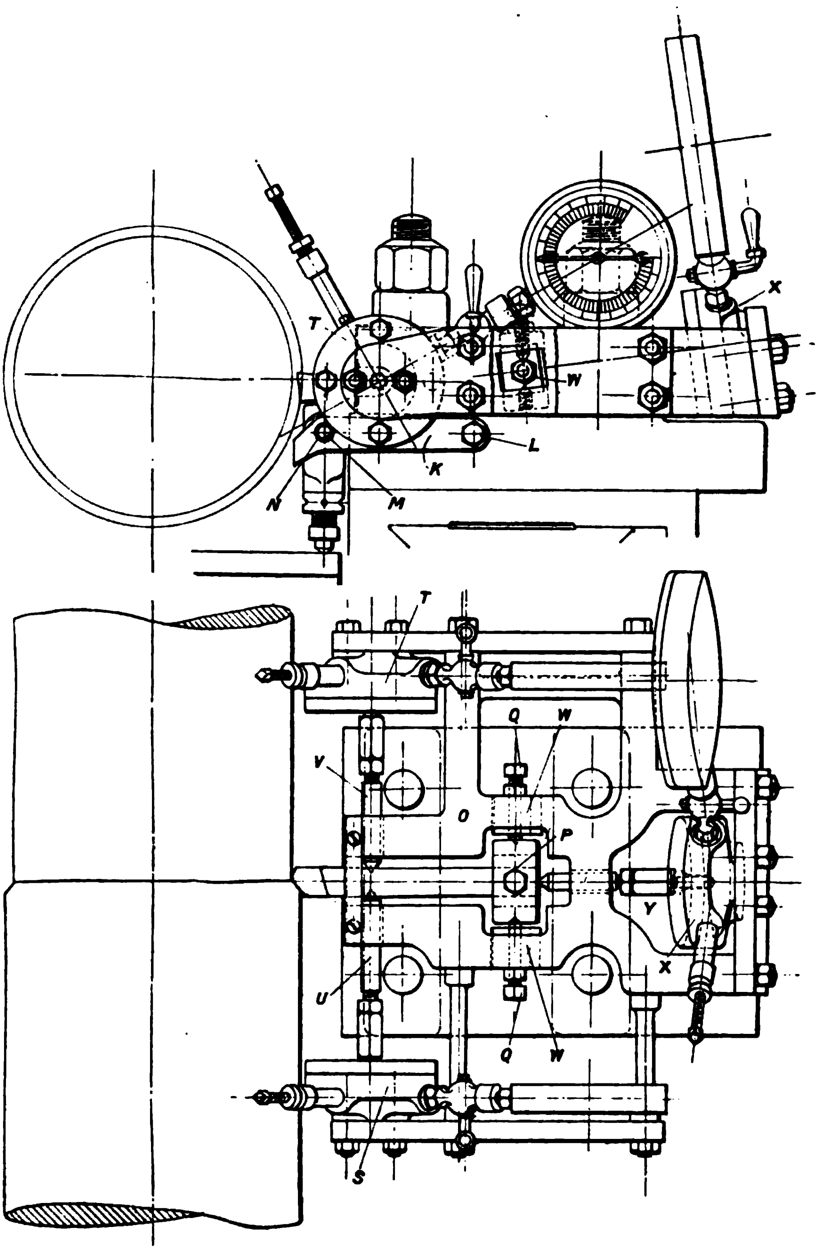

FIG. 6.—LATHE TOOL DYNAMOMETER—SIDE ELEVATION AND PLAN.

popmarks in the bottom and top of the tool itself. Thus tool and gimbal together are movable about the horizontal axis QQ; whilst the tool alone is free to move about the (nearly) vertical axis formed by screws RR.

There is but little force exerted upon, and therefore but little frictional

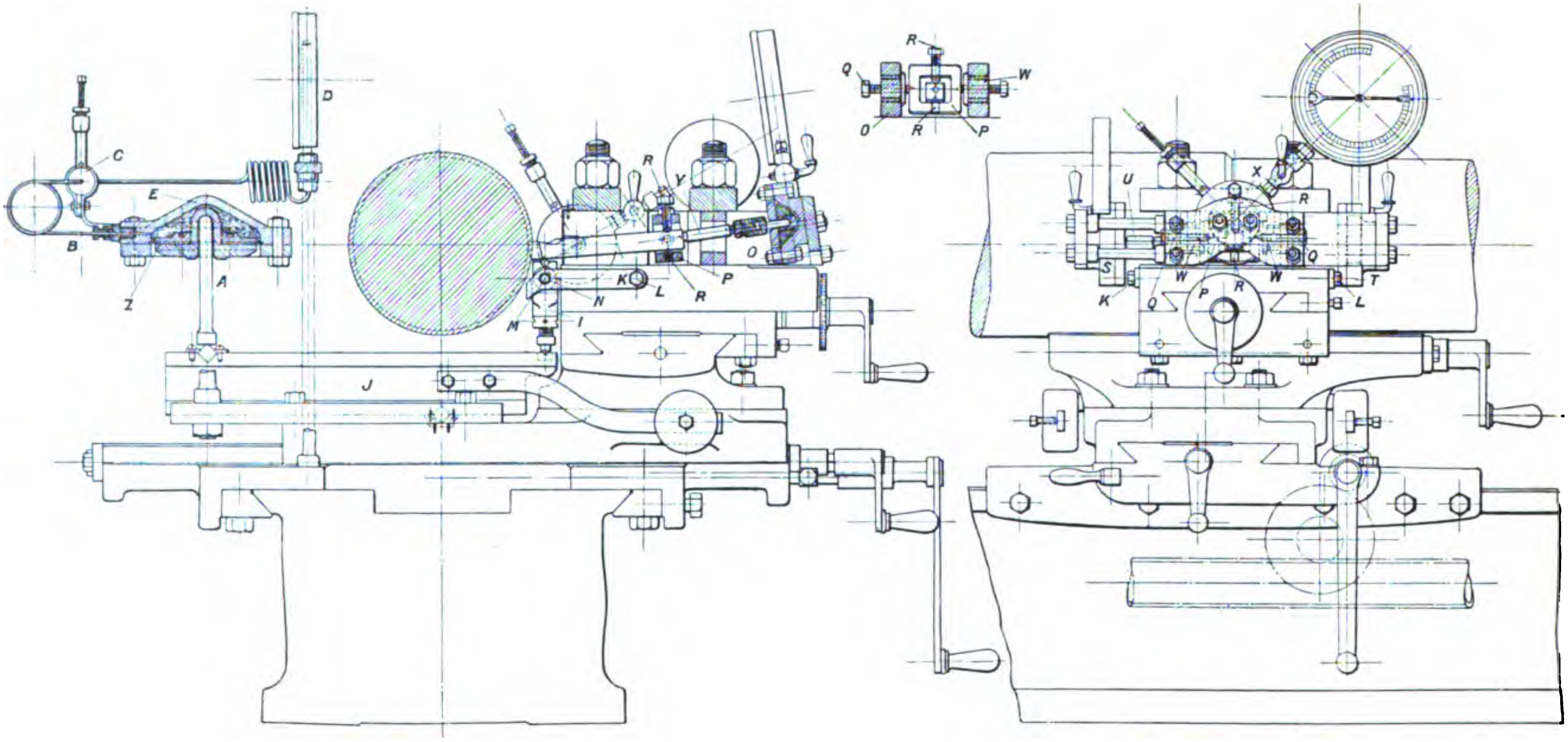

FIG. 7.—LATHE TOOL DYNAMOMETER—SIDE AND END ELEVATIONS.

resistance due to, the points of screws QQ—as also RR—for the cutting force, acting vertically on the tool point, is almost wholly balanced by that on the strut M, which transmits the thrust to another strut A, and so into the diaphragm pressure measuring vessel E. A and M might be in one piece if there were room; but to get round the bar a cast-steel beam J is interposed instead, which has knife-edged fulcrums. The actual force on the tool, corresponding to a given pressure shown by gauge D, was ascertained by hanging known dead weights on the tool point. In this way a direct calibration is effected without having to rely upon measurements of leverages or areas, and the vertical component of the cutting force V is accurately known.

The component parallel to the axis of the work, called the traversing force T, is found by placing two struts U and V—similar to M—between the sides of the tool and the two diaphragm force measures S and T—see plan. In order to avoid lateral oscillations it is necessary to put an initial pressure in the struts U and V, and on the vessels S and T, by screwing out the nuts and lengthening the struts before commencing to cut. If, then, the diaphragms are properly adjusted, the pressure in S falls by as much as that in T rises when the cut goes on. If, upon starting to cut, S is observed to rise and T to fall, the traversing force is said to be negative ($-T$). This occasionally happens.

Lastly, the approximately horizontal component of the cutting force is measured by direct transmission through strut Y to diaphragm pressure vessel X.

Before taking a cut a pressure is put on this diaphragm also by screwing out Y, and this thrusts the blocks W against the front of their slots in the casting O, but not against the work.

The backward or surfacing force S occasioned by the tool taking its cut first relieves the blocks from thrust against O, and, if care is taken not to put too great an initial pressure on the diaphragm, the whole value of the force S will finally be indicated by the pressure in X. The axis of rotation of the tool QQ and RR are close to the end into which strut Y bears, so that the small motions of the tool point do not affect the direction of V and the reading of X.

For further details regarding the construction of the diaphragm vessels and the mode of using the dynamometers during a test, reference must be made to the above-mentioned paper.

CHAPTER IV

CUTTING FORCES—RESULTS

MORE than one thousand trials on different materials with various shapes and sizes of cut, and with differently formed tools, have been carried out by the authors with the dynamometers above described. A first report, dealing with trials made up to April 1904, was read by one of them at the Chicago meeting of the Institution of Mechanical Engineers. A second report, in which the mass of data accumulated by the other author as the result of about six hundred trials, made partly in the course of laboratory work with students as well as by special research, is now in preparation.

The accompanying Figures, Nos. 8 and 9, give a general abstract in pictorial form of these new results. They allow of certain rough generalizations being made which we require for the purposes here in view.

Fig. 8 refers to experiments made upon Whitworth fluid-pressed steel of medium hardness, and Fig. 9 to those made upon medium-hard cast iron. Both bars were presented to the School of Technology for the purpose of these experiments by Messrs. Sir W. G. Armstrong, Whitworth & Co. Their compositions are closely similar to those of the corresponding bars made use of by the Manchester Committee in 1902.

In both figures the vertical cutting pressures (expressed in tons per square inch) are plotted on two bases: the first being the area of the cut; the second the cutting angle of the tool. (Here the cutting angle means the "included angle" plus the front clearance.) These pressures are represented by the full line curves. The broken line curves give the traversing and surfacing pressures expressed as percentages of the vertical cutting pressures.

Steel.—We see that, with steel, the vertical cutting pressure diminishes as the area of section of the cut increases, whatever the tool angle may be. The variation is numerically exhibited in the following table:—

TABLE IV

Medium Steel cutting Pressures: Tons per Square Inch.

Area of section of cut . .	0·005 sq. in.	0·01 sq. in.	0·02 sq. in.	0·04 sq. in.	0·06 sq. in.
Size of lathe corresponding to this area of cut	6 in.	8 in.	12 in.	16 in.	20 in.
Cutting angle of tool 80°	112	108	103	98	94
„ „ 70°	104	100	96	93	91
„ „ 60°	92½	90	87½	84	80

Thus for tools of the usual workshop angle for steel, say 70°, we find that the actual cutting pressure is about 105 tons per square inch for small lathes, diminishing to about 90 tons for the larger sizes. The sizes of lathe corresponding roughly to the various areas of cut are given in the second line of the table.

The traversing pressure, being that force per square inch area of cut which is required to hold the tool up to the work lengthwise of the bed, diminishes from 21 to 15 per cent. of the vertical cutting pressure as the area of the cut increases from 0·005 to 0·06 square inches when the tool angle is 70°. The traversing force varies therefore from about 22 tons to about 14 tons per square inch as the lathe centres increase from 6 in. to 20 in. The surfacing pressure, being that force per square inch of area of cut which is required to

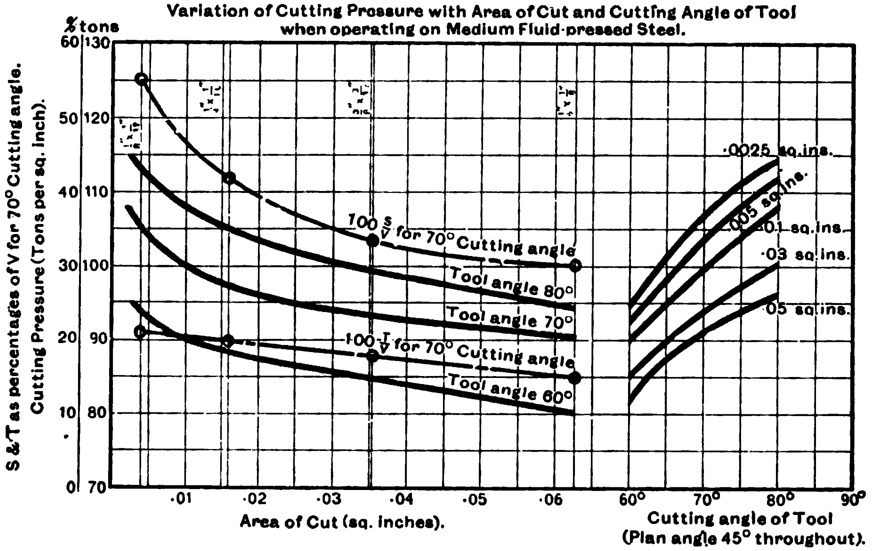

FIG. 8.

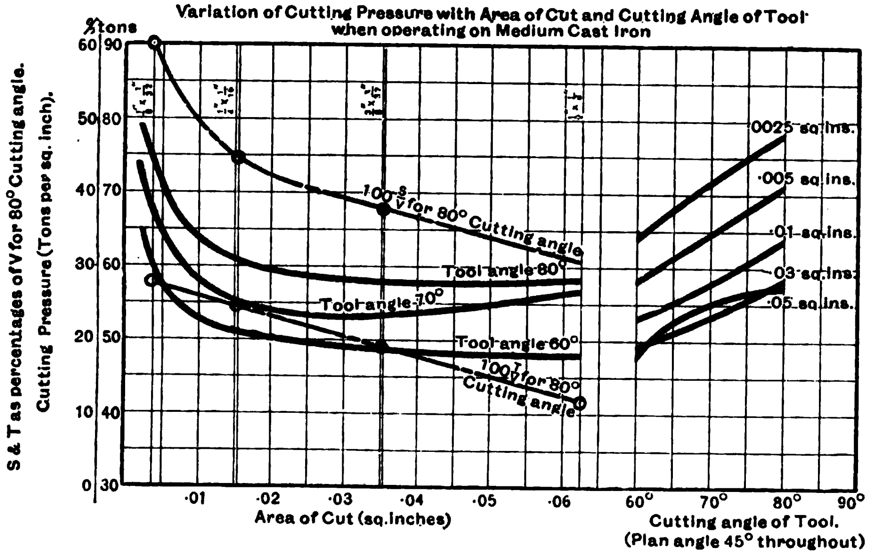

FIG. 9.

hold the tool up to the work in a direction across the bed, also diminishes as the sectional area of the cut increases. Its value is 53 per cent. of the vertical pressure, or 55 tons per square inch for small lathes; and it diminishes to about 30 per cent., or 27 tons per square inch for the larger sizes.

On the right-hand side of the figure the laws of variation of the cutting pressure with the cutting angle of the tool are represented for various areas of cut. They are seen to follow approximately linear laws, which differ with the various areas of cut.

Cast Iron.—In the case of cast iron the results are, generally speaking, of a similar character; the vertical pressure showing, however, a tendency to increase again for large cuts after passing through a minimum.

The variations of the vertical cutting pressure is set forth in the following table, numerically:—

TABLE V

Medium-hard Cast Iron cutting Pressure: Tons per Square Inch.

Area of section of cut . .	0·005 sq. in.	0·01 sq. in.	0·02 sq. in.	0·04 sq. in.	0·06 sq. in.
Size of lathe corresponding to this area of cut	6 in.	8 in.	12 in.	16 in.	20 in.
Cutting angle of tool 80°	72	64	59	57½	58
" " 70°	64	58	54	53	57
" " 60°	57	53	50	48	47

The traversing pressure diminishes from 28 to 12 per cent. of the vertical pressure as the area increases from 0·005 to 0·06; so that for small lathes the traversing pressure has a value of about 20 tons per square inch for an 80° tool; and about 7 tons per square inch for the same tool angle for large sizes of lathe.

The surfacing pressure varies from 58 to 31 per cent. of the vertical as the area increases; and may be taken at 42 tons per square inch for the smaller, and 18 tons per square inch for the larger sizes of lathe; the tool angle being about 80° in both cases.

The variations of the cutting pressure with the cutting angle of the tool are shown on the right side of the figure. They may be said to follow roughly linear laws, as for medium steel.

The laws of the variation of the cutting pressures when machining steel or cast iron in a lathe or planing machine are apparently very involved. Their detailed discussion in the light of the experiments made by both of the authors is reserved pending further investigation.

The results here given are in agreement with those of other careful experimenters; but it may be pointed out that no other method than that by means of hydraulic supports, having such areas and proportions as to ensure *direct measurements* of the forces and a non-vibrating tool, can be held to be comparable for accuracy with those here described.

In the sequel, we have adopted for the purposes of design, a cutting pressure of 115 tons per square inch as a value of the vertical force to cover all cases when machining steel. This is on the safe side from the point of view of strength and stiffness. It is also intended to allow for the fact that the work per minute required for cutting is greater than the product of the cutting speed v, and the vertical cutting pressure V, by the work done in pushing away the shaving over the face of the tool against the force of friction. To obtain the value of this additional work we proceed as follows :—

Referring to Figs. 10 and 11[1] it may be premised that the cutting moves off in a plane perpendicular to the cutting edge—except in the case of very light cuts—whereas the horizontal component H of the resultant cutting force R makes a certain angle δ with that plane. In order, therefore, to obtain the work done in pushing the shaving back, we must take the product of the speed of motion of the shaving over the tool, and of the component of H in the direction of that motion. If s be the shaving's speed, we have, in symbols, for this work—

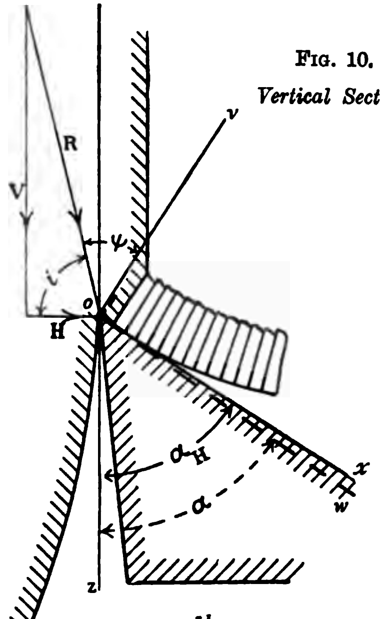

FIG. 10.

Vertical Section.

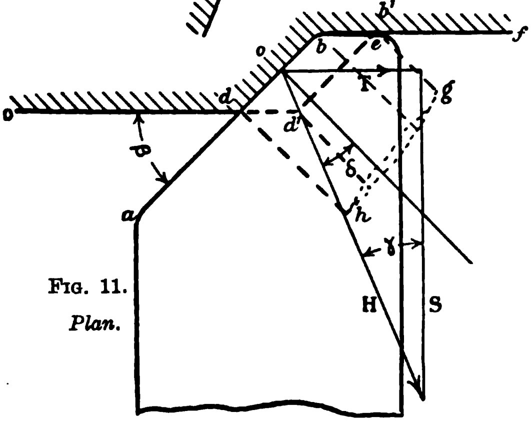

FIG. 11.

Plan.

$$sH \cos \delta \cos (90 \text{ deg} - a) = sH \sin a \cos \delta$$

Now s bears to the cutting speed v the same ratio which the length of the shaving bears to that of the surface from which it was cut; and this has been found by experiments to be about 0·45 for steel, so that $s = 0{\cdot}45\ v$.

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[1] Reproduced by permission from *Proc. Inst. Mech. Eng.*, June, 1904.

Also $H = \frac{V}{\tan i}$. If W be written for the whole cutting work, we have therefore—

$$W = vV + s\frac{V}{\tan i}.\sin a \cos \delta,$$

$$\text{or } W = vV\left[1 + 0{\cdot}45\frac{\sin a \cos \delta}{\tan i}\right]$$

For a 70° tool, $a = 70$ deg., $\delta = 16$ deg., $i = 70$ deg. (*vide* Fig. 29),[1]

$$\therefore W = vV\left(1 + 0{\cdot}45\frac{0{\cdot}9397 \times 0{\cdot}9613}{2{\cdot}7475}\right)$$

$$= 1{\cdot}13vV$$

That is, the backward pushing of the shaving makes an addition of 13 per cent. to the cutting work, as estimated by multiplying the cutting speed by the vertical cutting force.

Thus, taking 96 tons as the vertical cutting pressure (*vide* table above for medium steel), we must multiply this figure by 1·13, obtaining 108·5 tons for the value of the vertical force, which multiplied by the cutting speed will give the real cutting horse-power.

In addition, a margin must be allowed, when estimating the total motive power required to drive a lathe, for the frictional losses in the. headstock gearing and bearings.

An addition of another item per cent. on this account brings the gross cutting pressure up to the figure of 115 tons per square inch given above.

For the value of the surfacing pressure we have taken 50 tons per square inch; and for that of the traversing (or feed) pressure we have taken 25 tons per square inch.

These values are considered by us sufficient to provide a reasonable margin for all workshop contingencies so far as they ought to be foreseen when designing the feed mechanism for strength and stiffness. The results obtained also by the use of these values in our calculated designs show a good agreement with the proportions adopted in practice.

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[1] *Loc. cit.*

CHAPTER V

DURABILITY OF HIGH-HEAT STEEL TOOLS—VARIATION OF CUTTING SPEED WITH ANGLE OF TOOL AND WITH SIZE AND SHAPE OF CUT

In order to understand the nature of the phenomena occurring during the failure of a turning tool, a close study of the mode of formation of the chip was undertaken.

To facilitate matters in this formidable research it was first arranged to take a cut at a very slow speed, so that the action might be observed at leisure.

The lathe was driven by taking several turns of a wire rope round one of the cone pulleys and then leading it to a hand winch. When the labourer turned the winch handle, the cutting speed was 1 ft. in from one to five hours, according to the number of back gears in use. Lines having been ruled on the uncut surfaces of the work, forming squares of $\frac{1}{60}$ in. side, the distortion of these squares during and after the cut could be easily studied.

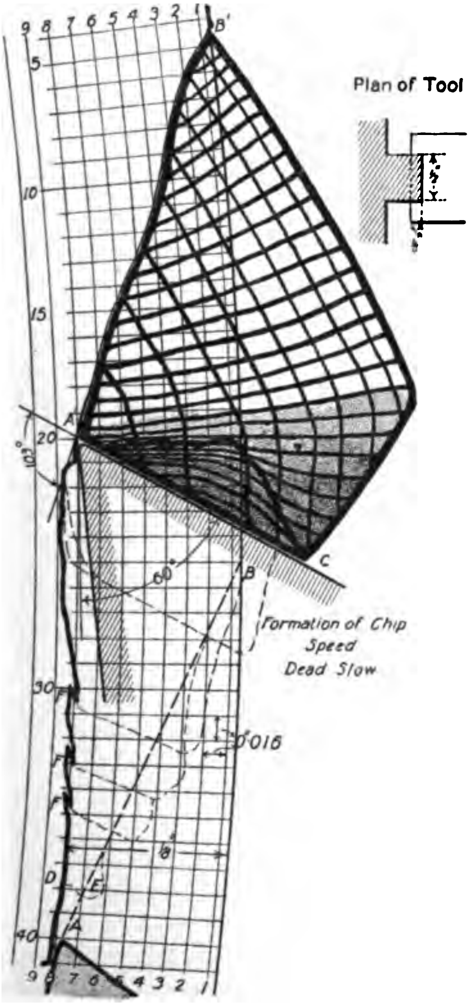

Fig. 12.

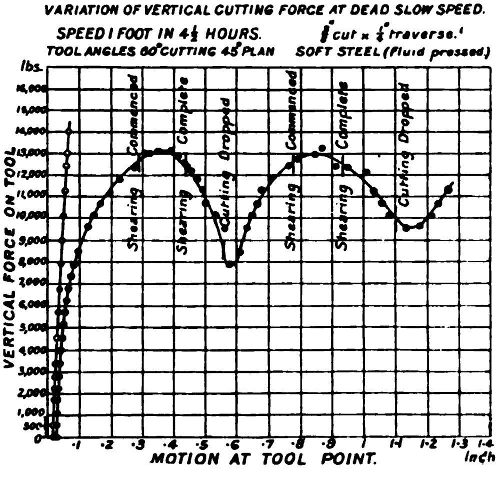

Fig. 13.

A chip formed by taking a cut $\frac{1}{8}$ in. deep from a collar $\frac{1}{2}$ in. wide, with a tool square-nosed in plan and having a 60° cutting angle, is depicted in Fig. 12. The originally circumferential and radial scribed lines are also seen, and the forms these lines take in a medium hard steel chip have been drawn to a scale about seven times full size.

At such dead-slow speeds the shaving breaks off in a succession of chips, which do not at all adhere to each other, by shearing off at an inclination of about 15° to the vertical along AB or A^1B^1. This angle does not vary with the angle of the tool, but the angle B^1A^1C of the chip becomes greater or smaller according as the tool is keener or more obtuse. The line AB shows where a chip has just been shorn off and has dropped away; and the tool is seen beginning to act upon the wedge-shaped portion BAA^1. However sharp the tool may be, its edge is somewhat rounded; and it is, therefore, difficult for it at first to get below the surface of the work. Thus we find that the "surfacing" force—or the horizontal force pushing the tool forward square to the lathe bed—is greater at this time than at any other, although the vertical force is then least, as the last chip has just been cleared off. If the surface of the bar is examined afterwards, it is found to have a burnished appearance just at this part, AD, where the tool pressed hard upon it. Once the tool enters below the surface it crushes up the material in front of it, and this is caused to flow outwards both along and across the tool surface, as at DE. Slipping of this material soon takes place over the tool and away from the work; but this slipping is not continuous, taking place intermittently and at longer intervals the more the tool cuts into the wedge BAA^1. Each slip is accompanied by a small tear or crack running in front of the cutting edge, as seen at F; but these cracks extend for but a short distance at first, being healed up by the radial compression produced by the friction of the crushed material on the top of the tool when it comes to rest after having made a "slip." The same action is seen in the crushing of a short ductile specimen in the testing machine; the friction between test piece and platen being powerful enough to prevent cross-extension, uniform throughout its length, from taking place; so giving the crushed piece the barrelled shape commonly observed.

These little cracks—FF—run deeper, although at longer intervals, as the crushed material spreads further back over the top of the tool, and the consequent "slips" become more violent, until one such tear or crack extends right up to A^1, and so out by A^1B^1, and the chip drops.

The action at high speed is in many respects quite different from what takes place at "dead slow." The successive main shears—or, rather, slides—of the shaving take place at shorter space intervals, for the slipping of the shaving over the tool now takes place continuously, and the cracks are no longer healed up by compression. The tear from each of the cracks is now continued right out to the surface of the shaving. The progression of this tear across the shaving is not instantaneous, however, and by the time it has reached the upper surface the chip has advanced some distance from the point of the tool. By reference to Fig. 13, which shows the variation of the force during chip formation at slow speed, it will be found that the vertical cutting force is greatest between the moment when the tear starts and that when it reaches the surface.

If we now refer to Fig. 14, which shows how the shaving is formed at high speed, we see that this region of maximum stress extends from near the point to a considerable distance backwards on the surface of the tool. The tear at the very point keeps running ahead of the tool, and is due, not to the direct action of that point, but to the twisting of the material from behind the point where the pressure on the tool is greatest. In fact, the point itself is protected from all action of the work upon it by a small heap of the cut material being collected in the little space opened out by the advancing tear, and resting on a shelf of uninjured tool surface extending from the very edge of the tool to a greater or less distance back therefrom. This heap is firmly adherent, as to its base, to the shelf, but as to its upper part, is continually being removed and replenished from material left on the surface of the bar by the irregularity of the advancing tear. The nose of the tool sustaining the heap only cleans up the shaft, but does not cut the main body of the material from it.

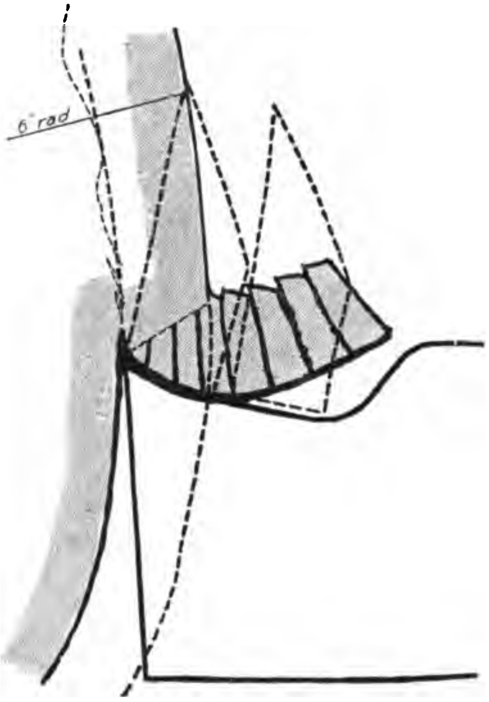

Fig. 14.

The shaving, at that part of it where the crack is just running across, presses hard, as we have seen above, upon the top of the tool, and by reason of the speed with which it runs over the surface and the high temperature which it imparts to it, gradually wears a groove therein. This grooving starts at some distance back from the cutting edge and gradually advances by wearing away the shelf of unworn tool and diminishing the size of the heap, until, when the groove reaches the cutting edge and the shelf is all gone, the tool "fails." Failure is the process—which occurs almost instantaneously after the groove intersects the nose of the tool and produces a feather-edge there—of blunting and destroying this edge, the depth of cut then dropping considerably from its proper value. The "work of bluntness" is thereby largely increased, generates a corresponding larger amount of heat right at the edge of the tool, and the cutting power is immediately destroyed.

From a consideration of the facts above discussed it will be understood that the durability of a tool depends upon the length of time required for the groove scooped out by the shaving on the top surface of the tool to reach the cutting edge. The laws of the rate of this grooving have not, so far, been definitely established; but the rate at which the abrasion of the tool top takes place would appear to depend upon—(1) The normal pressure on the tool due to the twisting of the shaving. This is independent of the area of the cut—at least, to a first approximation, but increases with the cutting angle, the law of this latter variation having been given above (Figs. 8 and 9). (2) Upon the rate at which the shaving slides over the tool. For a given size and shape of cut this is a constant fraction of the cutting speed. But it varies with the cutting area and the shape of the cut, being smaller the heavier the cut and the greater the "crowding." (3) Upon the temperature of the top surface of the tool. This temperature is that of the part of the shaving with which it is in contact. The shaving is hottest where the twisting or shearing is greatest—that is, in the region where the crack is running across the shaving, and not at the point of the tool, where shearing is just commencing. Therefore, the tool is found to be cooler at the cutting edge than at the part where the groove is forming. Heat flows, of course, by conduction from the hot region behind the point to the cutting edge; but the edge is kept cool by a continuous supply of fresh cold bar. Thus the tool does not fail at high speeds because the cutting edge gets too hot, but because the tool gets hot some distance back from the edge, and so facilitates the grooving action above described, which leads to its destruction by finally extending to the edge.

We are now able to see why it is that acute angled tools, although the cutting force on them is much less than upon obtuse tools, do not last so long as those of a medium keenness. When the tool is very keen-angled, there is but little material to be abraded before the groove reaches the cutting edge. With tools having such a large cutting angle as 90°, on the other hand, there is a relatively large amount of tool steel to be worn away before the groove works back to the cutting edge; but the cutting stress is so great, and the amount of twisting of the shaving, *i.e.* the temperature of shaving and tool is so high, that abrasion proceeds with great rapidity, and the tool fails just as quickly as a very keen one.

The most durable angle is an intermediate one, say, from 65° to 75°, for which the cutting stress and the temperature are not so high, whilst the amount of material to be abraded before failure is moderately large.

There is little doubt that when the laws of the variation of the temperature of shaving and tool with different cutting angles, sizes, and shapes of cut, and of the rate of abrasion as depending on normal pressure, speed of rubbing, and temperature, are definitely determined, it will be possible to indicate how the tool should be ground in order to meet with the best efficiency, both as to power and durability, the very various conditions to be found in practice.

Durability Experiments

The experiments of the Manchester Committee give a large amount of information regarding the durability of the new tool steel, as depending on the speeds at which cuts of varying areas are taken. Figs. 2 and 3, *supra*, were, in fact, different curves of constant durability.

The most exhaustive series of experiments so far made on the economy and durability of cutting tools for the lathe are, however, undoubtedly those of Mr. F. W. Taylor. They will be found described in the voluminous paper presented by him at the New York meeting of December 4 to 7, 1906, of the American Society of Mechanical Engineers.

With an ample supply both of material to be cut and of tool steel to cut it, he was able to make a large number of experiments upon the cutting speeds of both low- and high-heat steels under different conditions; as to shape of tool, shape and size of cut, and nature of material; the tools being run at such speeds as permitted of the tests lasting for about twenty minutes each.

When comparing his results with those of the Manchester Committee, in the above paper, he expressed the opinion that the importance of cuts having a high proportion of width to thickness in increasing the life of the tool had been overlooked by the Manchester Committee; and that their conclusion that the "proof" cutting speed (that speed, viz., at which a tool may be expected to last just twenty minutes) was a function of the area only, and not of the shape of the cut, was not only not well founded, but that "any formula or even any data containing the area of the cut, or shaving, as a single element is valueless from a scientific standpoint." One of the present authors replied to this criticism,[1] and an extract from his remarks is here given.

"A careful study of the author's results, as given in his Tables 70 and 76, Folder 11—which tables he cites as examples of the most successful series of experiments he has made—leads to the conclusion that, so far from revealing a glaring discrepancy from the law of cutting speed and area, enunciated by the Manchester Committee, the author's results do not differ from that law by more than 1 or 2 per cent. on the average throughout the range of depths of cut and widths of feed ordinarily used for roughing out. The author has given no experimental figures either in Tables 70, 76, or elsewhere, to substantiate the statement made in section 281 C, that the cutting speed may be $3\frac{1}{2}$ times as great when the shaving is $\frac{1}{64}$ in. thick, as it is when the shaving is $\frac{3}{16}$ in. thick. The finest feed in Table 70 is $\frac{1}{16}$ in. To elucidate the matter, I have plotted the 'standard' cutting speeds, as given by the author—and their corrected values as they should, in my opinion, be taken from Mr. Taylor's own results—upon a base of area of cut, *vide* Fig. 15. I have then drawn a fair curve through all the spots

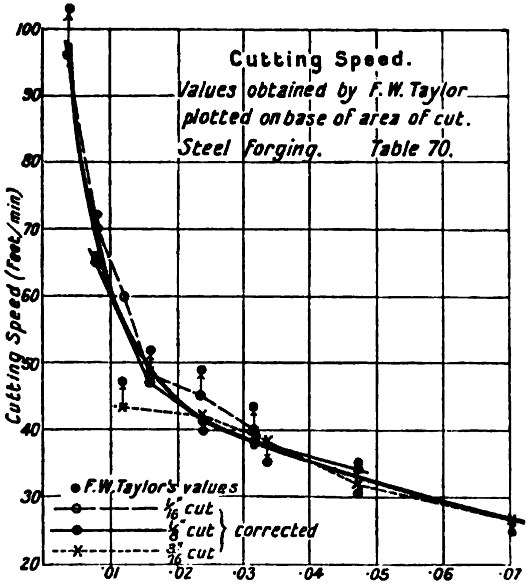

Fig. 15.

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[1] *The Engineer*, February 22 (1907), and *Proc. American Soc. Mech. Engs.* (1907).

on the diagram. An inspection of the way the author's results lie about and around this curve gives good ground for saying that Mr. Taylor's figures substantiate in a remarkable manner the fundamental law that the standard cutting speed with high-heat steel for a given material depends only upon the area of the cut, and hardly at all upon the depth of that cut or the width of feed. The author has tabulated no results which show a serious divergence from this law, and until he produces something tangible in this direction, the statement he has made in section 281 C must be characterized as entirely misleading. If the author had plotted his results on a base of areas instead of as contour lines of constant feed upon a base of depth of cut, he might not have been led to make statements of this kind, so patently contradicted by the figures he himself gives. As a matter of fact, a simple formula of the form $V = \frac{K}{a + Bc^n}$ can be obtained to cover, not only the Manchester and Berlin, but also Mr. Taylor's results; with a divergence of only ± 5 per cent. This formula involves only the area a of the cut in square inches, the amount of the carbon c in the steel forging expressed as a percentage, and the constants K—depending on the tool steel—B and n—depending on the other chemical constituents of the bar—and makes no reference to either the thickness of shaving or depth of cut.

"It may not, perhaps, possess the accuracy of the author's formula in representing his own results—*vide* paragraph 770. Of this I have made no trial, but I have hopes of its greater practical usefulness as involving eight fewer constants, and only one variable for a given material. If the author is in possession of experimentally obtained figures, proving that there is a considerable divergence from the speed on area law for such fine feeds as $\frac{1}{32}$ in. and $\frac{1}{64}$ in., it may still be pointed out that he has himself expressly characterized—*vide* section 218—such light cuts as worthless for the determination of the laws of cutting speed.

"I am prepared to go a step further even than this, upon re-examination of the author's paper in its final form, so far at least as refers to the effect of the thickness of the shaving on the cutting speed—see paragraphs 291 to 306. In paragraph 297 the author gives the formula $V = \frac{1{\cdot}54}{t^{\frac{2}{3}}}$ for the standard speed of cutting (V), as dependent upon the thickness of the shaving (t). It appears, however—*vide* Folder 16—that the shaving in the experiments upon which this law is based were all 1 in. wide. The thickness of the shaving in inches is therefore the same as its area in square inches, so that the above formula might equally correctly have been written $V = \frac{1{\cdot}54}{a^{\frac{2}{3}}}$, where a is the area of the shaving, or the product of its breadth by its thickness. We have, therefore, no evidence from these experiments that the increase of the cutting speed as the shaving got thinner was due to the thinning. "It may just as logically be said that it was due to the diminution of area. Similar remarks apply to the formula of paragraph 302, in which with equal propriety the area might have been substituted for the length of the shaving."

As a further confirmation of the contention that the cutting speed may be expressed (except for abnormally thin shavings) as a function of the area of the cut only, without reference to its shape, we may refer to an able analysis of Mr. Taylor's paper made by Prof. Georg Lindner in the *Zeitschrift des Vereins* for July 6, 1907. Referring to Taylor's very complicated formula for the cutting speed as dependent on the shape of the cut (*vide* "Art of Cutting Metals," sections 767–814), Lindner gives simple expressions which approximately represent Taylor's results.

For soft steel he thus obtains—

$$V_o = \frac{K}{C}\left(\frac{C}{t}\right)^e$$

Now, for a tool 2 in. deep × $1\frac{1}{4}$ in. wide, the value of e is 0·5; so that the above expression becomes

$$V_o = \frac{K}{(Ct)^{0 \cdot 5}} = \frac{K}{\sqrt{a}}$$

In words: the "proof" cutting speed is, *according to Taylor's own results,* inversely proportional to the square root of the *area,* and does not at all depend on the depth of cut or the width of feed!

We consider, therefore, that we have ample ground for maintaining that the cutting speed is a function of one single variable only—viz. *the area of the cut;* and that the proportion of its breadth to its thickness is, for all ordinary cuts, of quite secondary importance.

CHAPTER VI

CUTTING SPEEDS FOR WORKSHOP USE

We have to thank Mr. F. W. Taylor, on the other hand, for a law of very great importance which he has deduced from his experiments. In the Manchester Committee's trials, the experiments were mostly made at speeds under which the tools lasted for twenty minutes. A few trials only were made which lasted two hours. The question which arises, and which is one of great practical importance, is: What diminution from the proof speed (*i.e.* the twenty-minute failing speed) must be made in order to enable the tool to last, on the average, for any given longer time?

Mr. Taylor's answer is: *The speeds vary inversely as the eighth roots of the durations of the runs.*

That is

$$\frac{V}{V_o} = \left(\frac{T_o}{T}\right)^{\frac{1}{8}}$$

$$V = (T_o^{\frac{1}{8}} V_o)\frac{1}{T^{\frac{1}{8}}}$$

$$= \frac{K}{T^{\frac{1}{8}}}$$

Here V is the required cutting speed to last T hours. V_o is the proof speed, or speed on which the tool lasted $\frac{1}{3}$ hour.

The practical usefulness of this law consists in this: If, by experiments such as those of the Manchester Committee, or of Mr. F. W. Taylor, we have found the "proof speed" of the steel (that is, the cutting speed which will enable it to last just twenty minutes) on a given material, we are in a position by employing this law to determine the proper cutting speed at which the tool must run if it is to last for any other (longer or shorter) time on that material.

We have to decide, for example, in discussing the rules for lathe design, what should be the standard cutting speed for each size of general-purpose (high-speed steel) lathe when that lathe is taking the cut of standard area suitable to its capacity (*vide* Chap. VII.).

It will be admitted that the larger the lathe the longer the tool should last before it is reground. Thus, whilst for a lathe of 6- or 8-in. centres the tool need not be expected to last for more than twenty minutes on the average, in the case of a lathe of 30-in. centres, it would appear to be desirable that the speed of cutting should be chosen of such a value as to allow the tool to run for a whole shift of three hours' duration. Knowing, then, the proof speeds for the various areas of cut which correspond to all sizes of lathe, all we have to do is to apply Taylor's law to these numbers in order to deduce the appropriate or standard cutting speed for each size. The results of such a series of calculations are given in lines 1 to 5 of the annexed table, and are shown in diagrammatic form by Fig. 16.

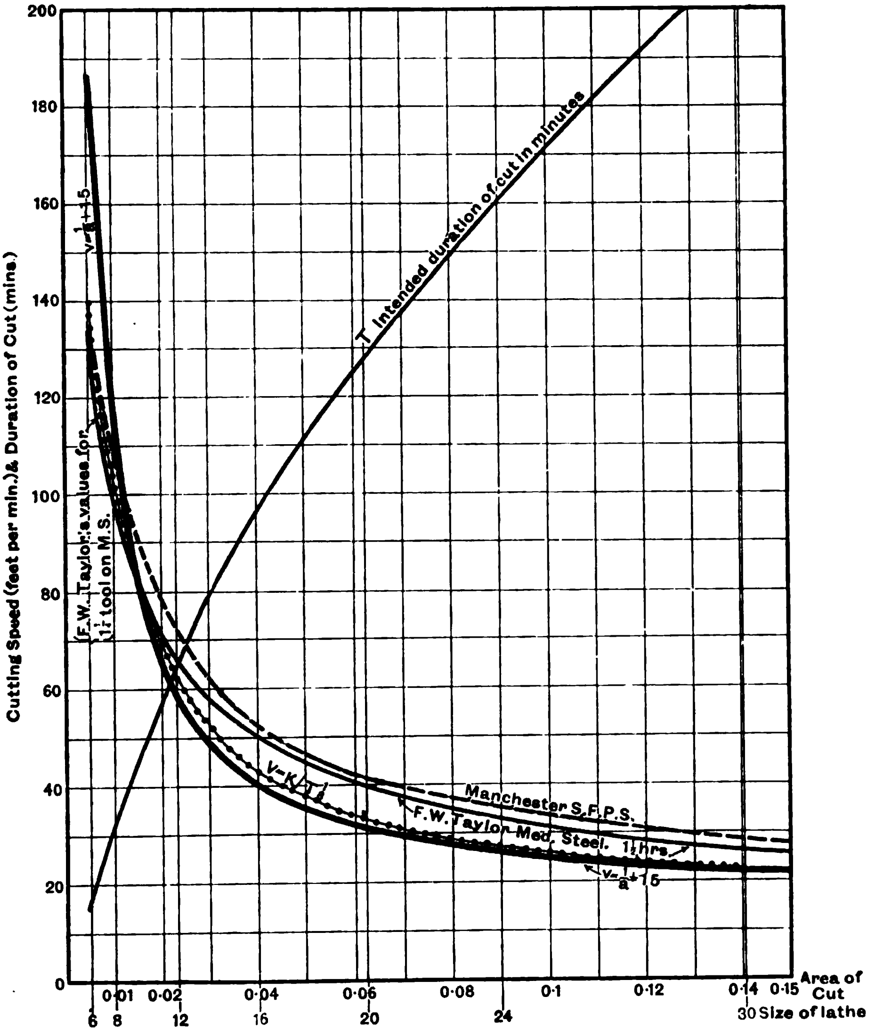


Fig. 16.

TABLE VI

1	Height of centres	6 in.	8 in.	12 in.	16 in.	20 in.	24 in.	30 in.
2	Corresponding area of cut . .	0·00564	0·01	0·023	0·04	0·0625	0·09	0·1405
3	"Proof speed" on S.F.P.S. . .	132	108	72·5	53	41·5	34·5	28
4	Proposed duration of cut (in mins.)	16	32	64	96	128	160	208
5	Corresponding cutting speed .	140	102·4	63·2	42·8	32·5	27	22·4
6	Standard cutting speed $\left(\frac{1}{a}+15\right)$	187	115	58·5	40	31	26·1	21·9
7	F. W. Taylor's speeds on medium steel for 1½ in. × 2 in. tool to last 1½ hours . . .	134	100	66	50	40	33·3	26·7

In line 6 we have further placed the speeds, as determined by means of the formula $v = \frac{1}{a} + 15$, which we decided upon for use in our lathe calculations before the above law of variation of cutting speed with duration of cut was available. It will be observed that our rule gives a sufficiently close approximation to the correct values of line 5, and that it may therefore be relied upon in the case of lathes to be used in general engineering work. Mr. F. W. Taylor's results, as taken from his paper (Folder 24, Table 143) for a 1¼-in. by 2-in. tool, on medium steel, have also been added for comparison, his intended duration of cut being 1½ hours for all sizes of lathes. His medium hard steel appears from these figures to have been not much more difficult to cut than the soft, fluid-pressed steel of the Manchester Committee.

The advisability of fixing the duration of the cut at the arbitrary figure of 90 minutes for all sizes of lathes, as Mr. Taylor has done, may very well be questioned; and we are of opinion that a gradation of the length of run to the size of the lathe is a more logical method, and will lead to a greater economy in working.

It may further be said with regard to Mr. Taylor's remarks that in Tables 144, 145, 146, 147, and 148 of the same folder he has given lists of cutting speeds for the various sizes of steel tool from 1¼ in. down to ½ in. He considers that every size of section has its own curve of cutting speed appropriate to a given length of run. The explanation offered being that small sections carry away the heat less easily than large ones, and that consequently a lower speed accompanies a lighter tool.

We are, however, of opinion that Mr. Taylor's actually observed lower proof speeds of the lighter tools was due not so much to difficulty with heat conduction as to the smallness of the radius of curvature adopted for the cutting edge at the front of the tool. Such small radii seem almost a necessary accompaniment to the rectangular shape of tool steel recommended by Mr. Taylor. With the square shape adopted in Britain, it is possible to have one and the same *large* radius of cutting edge for all but the smallest sizes of tool, and in that case our experience shows that *all sizes of tool*, between say ¾ in. to 1½ in. square, *may be given the same cutting speeds.*

The use of a square rather than of a narrow, deep section of tool is to be preferred in view of the very important fact that the nose of the former can be ground to a larger and more efficient radius than that of the latter. The shaving is in that way kept thin at the point of the tool, and the full depth of the cut can be maintained for a longer period.

The smithing of such square tools is also much easier, and less steel is wasted in reforging than in the case with Mr. Taylor's standard tool. There is no necessity when using square steel for the awkward side twist which Mr. Taylor has found to be necessary with the rectangular shape in order to keep the line of action of the resultant force within the base of the steel.

The weakening of the square tool by the slope of the top rake running into the scantling must, of course, be avoided by jumping the end so as to heap the material on the top. This is, however, a comparatively easy process. Such a shape also provides the material for a moderate number of regrindings. Our experience points to the deterioration of the tool and the necessity for its reforging and rehardening after it has been reground only a few times, and we are not in favour of the excessive depth which Mr. Taylor prescribes as the best form.

CHAPTER VII

LATHES FOR HIGH-SPEED CUTTING—STANDARD CUTS AND SPEEDS—RANGE OF SPEED—POWER

THE experimental results above enunciated are now to be made use of in the endeavour to frame rules for the design of lathes on rational principles. The experience of makers with lathes designed and made to work with the high-speed steel will be taken advantage of, so far as it is available, to check the formulæ so obtained; and it is believed that these rules will prove a useful guide in cases where such experience is entirely lacking.

The type of lathe which is to be considered is one for general purposes in an engineer's shop; and the rules are intended to cover a range of sizes from 6 in. to 6 ft., height of centres, in that type. Special lines of manufacture require special purpose lathes, and it is the truest economy to have such special lathes whenever it is possible. The conditions postulated, as to standard cut and ranges of speed, for general purpose lathes will then, of course, no longer hold; but the underlying principles are the same for all.

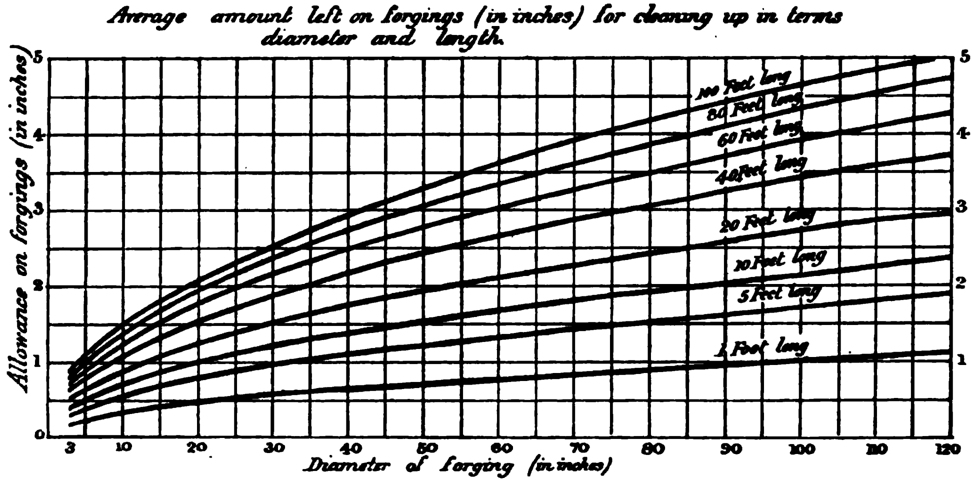


FIG. 17.

In the first place, it is necessary to fix upon the area of cut which ought properly to be taken upon a given material, say, mild steel, with each size of such general purpose lathe. It is obvious that a small lathe ought not to be expected to carry the same heavy cut as one two or three times its height of centres. For one thing, small castings and forgings show a smaller difference between their sizes, rough and finished, so that the amount to be machined is less than for large ones. The annexed Fig. 17 gives, in the form of curves, the averages of about a dozen lists of allowances on forgings sent in to us by different makers. The various curves are represented by a formula of the form

$$D - d = \frac{\sqrt{d}}{10} \times \sqrt[3]{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where D and *d* are the rough and finished diameters of the forging in inches; and L is its length in feet.

If, for example, we take the average diameter of the work equal to the height of lathe centres, and the length of the forging equal to 6*h*; then we have

$$D - d = \frac{\sqrt{h}}{10} \times \sqrt[3]{\frac{h}{2}} = \frac{h^{\frac{5}{6}}}{12{\cdot}6}$$

and the depth of cut would be

$$\frac{D - d}{2} = \frac{h^{\frac{5}{6}}}{25} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

So that the average depth of cut should increase almost in proportion to the size of the lathe.

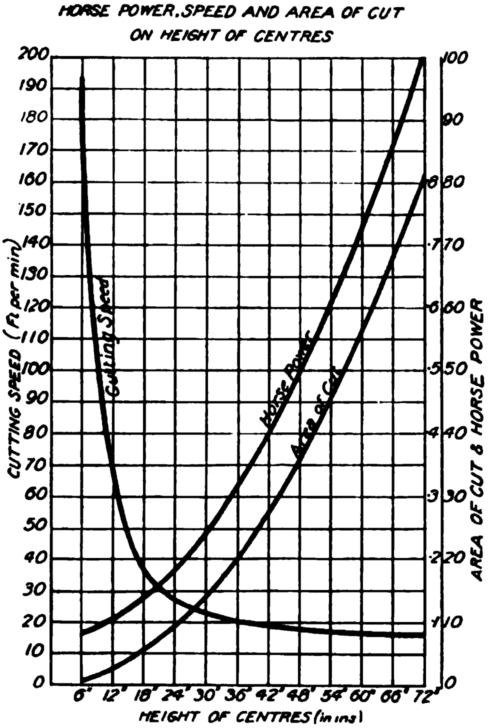


FIG. 18.—POWER, SPEED, AND CUT.

From other data obtained by the experience of a number of the leading firms of machine tool-makers we have been led to the conclusion that the area of cut for different sizes increases as the square of the height of centres; and the following general rules have been adopted in this work as meeting with a wide acceptation for the machining of mild steel forgings:

Let c = depth of cut in inches,
t = width of traverse feed in inches,
a = area of cut ($= ct$) in square inches,
h = height of lathe centres in inches.

$$\text{Then} \quad c = \frac{h}{40}, \; t = \frac{c}{4} = \frac{h}{160}$$

$$\text{and} \quad a = ct = \frac{h^2}{6400} \quad (27)$$

A curve representing this expression has been drawn on Fig. 18, the base being height of centres, the ordinates area of cut in square inches (right-hand scale).

Having thus determined the standard cut for which each size of lathe is to be designed, we are in a position to decide upon the horse-power to be supplied to it. From the area of cut and the cutting pressure we have the force to be overcome at the tool point; and the product of this into the cutting speed at which the standard cut is to be taken gives the work done in foot-tons per minute. Now the formula

$$v = \frac{1}{a} + 15 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

deduced from the Berlin and Manchester committees' experiments gives the proper speed, if a is the area of the standard cut.

In order to be on the safe side in making our calculations of strength and power, we shall take the cutting pressure as for steel, and at the high figure of 115 tons per square inch. This allows about 20 per cent. for the frictional loss between belt and tool and that due to the shaving. We thus write

$$\text{GHP} = \text{gross horse-power} = \frac{115 \times 2240}{33{,}000} av = 7{\cdot}8av$$

But $$a = \frac{h^2}{6400}, \text{ and } v = \frac{6400}{h^2} + 15$$

therefore $$av = \frac{h^2}{6400}\left(\frac{6400}{h^2} + 15\right) = 1 + \frac{h^2}{427}$$

and $$\text{GHP} = 7{\cdot}8\left(1 + \frac{h^2}{427}\right) = 7{\cdot}8 + \frac{h^2}{54{\cdot}8} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

The gross horse-power so calculated is given in the annexed table for a series of lathes with heights of centres from 6 in. up to 6 ft.

TABLE XXV

Standard Cuts and Gross Horse-power (for one tool) for sizes from 6 *in. to* 6 *ft.*

Height of centres	6 in.	12 in.	18 in.	24 in.	30 in.	36 in.	42 in.	48 in.	54 in.	60 in.	66 in.	72 in.
Depth of standard cut	0·15	0·30	0·45	0·60	0·75	0·90	1·05	1·20	1·35	1·5	1·65	1·80
Width of standard traverse	0·0375	0·075	0·1125	0·15	0·1875	0·225	0·2625	0·30	0·34	0·375	0·4125	0·45
Area of standard cut	0·0056	0·0225	0·052	0·09	0·14	0·203	0·276	0·36	0·46	0·56	0·68	0·81
Proper cutting speed	193	$59\frac{1}{2}$	$34\frac{1}{3}$	26	22	20	$18\frac{3}{5}$	$17\frac{3}{4}$	$17\frac{1}{4}$	$16\frac{3}{4}$	$16\frac{1}{2}$	$16\frac{1}{4}$
Gross horse-power	8·4	10·4	13·7	18·3	24	31·4	40	50	61	74	87	102

These quantities are also figured as the ordinates of the three curves of Fig. 18 on a base of height of centres. The first curve gives the value in square inches of the area of the standard cut—right-hand scale—for a lathe of any given height of centres; the second curve gives the cutting speed—left-hand scale—at which this cut should be capable of being taken on work of any diameter up to twice the height of centres; and the last gives the net horse-power required to carry this out.

Criticisms as to the propriety of these standard cuts, and consequent powers to be supplied for the various sizes of lathes, may be made, and will be welcomed by the authors. It should be pointed out, however, that it is only upon the question of the *size of the cut* which each size of lathe ought to be expected to take that there is any room for discussion or choice of opinion. Once the rule for that is agreed upon, all the rest follows as a matter of logical consequence from the experimental results now available. From the area of the cut follows the appropriate speed, having due regard to the life of the tool; the force required for taking the cut is also determined, and the required net horse-power follows at once.

GREATEST AND LEAST SPINDLE SPEEDS

Fundamental to the design of the fast headstock is a knowledge of the range of speed of the spindle and the number of speeds into which this range must be divided. Rules must be laid down for the greatest and least spindle speeds, and compared with practice, for all sizes of lathes; and the proper value of the constant geometric ratio of successive speeds must be considered.

Denote by—

N_g the greatest number of revolutions of the spindle.
N_l „ least „ „ „
n_g „ greatest „ „ cone.
n „ least „ „ „

The value of N_g, the *highest* spindle speed, will obviously depend on the smallest diameter of work which ought to be allowed in the given size of lathe and upon the highest cutting speed at which any cut is required to be taken on that diameter.

Now a diameter equal to one-eighth the height of centres ($h/8$) is about the smallest size of bar that is usually put into a lathe. For any size less than this the work would be more economically performed upon a smaller lathe. Also a cutting speed of 120 ft. per minute is found to be as high as is ever necessary on this small diameter for lathes from 6 in. to 18 in. height of centres. If the cut be of less area than that corresponding to this speed, it is a finishing cut, and must not be taken too fast for fear of vibration. Now if v represent this cutting speed in feet per minute, we have obviously

$$N_g = \frac{12v}{\pi h/8} = \frac{12 \times 120 \times 8}{\pi h} = \frac{3670}{h}$$

or, say,

$$N_g = \frac{3600}{h} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (30)$$

For lathes larger than 18 in. centres a cutting speed of 80 ft. per minute on the smallest diameter is found to be sufficient; and this gives a value

$$N_g = \frac{2400}{h} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

Speaking generally, it may be said that the cone runs upon the spindle and may be coupled directly to it for lathes up to 18 in. centres; whereas above this the cone is usually off the spindle, and drives it, even at the highest speeds, through a pair of wheels. Recently, the mounting of the cone —if any—upon the spindle has not been so generally resorted to except for lathes below 12-in. centres, and from the point of view of belt efficiency it is certainly much better that the spindle should never be directly coupled to the cone.

The smallest number of revolutions of the lathe spindle N_l is determined very simply by the consideration that it must be possible to take the standard area of cut for which the lathe is designed ($h^2/6400$) upon work of the face-plate diameter, and at the appropriate speed $\left(\frac{6400}{h^2} + 15\right)$ at which the tool can last for a reasonable time.

We thus obtain

$$N_l = \frac{12v}{\pi \times 2h} = \frac{6}{\pi}\frac{\left(\frac{6400}{h^2} + 15\right)}{h} = \frac{2}{h}\left(\frac{6400}{h^2} + 15\right)$$

or

$$N_l = \frac{12{,}800}{h^3} + \frac{30}{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

The values of N_g and N_l determined by these rules are entered in lines 1, 2, and 3, of Table XXVI., and have been plotted on Fig. 19.

The curves of greatest and least spindle speed thus rationally determined are seen to agree well, on the whole, with the most recent practice; for in Fig. 19 the dots showing values for lathes actually built by some of the best makers are found to cluster around the path of the curves. The break in the curve for N_g is due to the change made in the assumed value of the highest cutting speeds ever required on small diameters, viz. from 120 to 80. A region of indefiniteness occurs for sizes between 12 in. and 18 in. height of centres; and this is just what we ought to expect when, for these sizes, we sometimes have the cone mounted upon the spindle, and in other cases driving the latter through gearing, even for the highest speeds.

No such break occurs in the curve for N_l; as, for the lowest speeds, there is always at least one back gear interposed even for small lathes.

The ratio of greatest to least spindle speeds is, of course—

$$\frac{N_g}{N_l} = \frac{\left(\overset{\text{small}}{3600} \text{ to } \overset{\text{large}}{2400}\right)h^2}{12{,}800 + 30h^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

and these values are also tabulated in Table XXVI., lines 4 and 5, and plotted in Fig. 19.

We may fix the highest required spindle speed (N_g) by the consideration that every high-speed steel lathe should be capable of finish-boring and drilling holes as small as $h/16$—one-sixteenth of the height of centres—in diameter with rapid-cutting steel drills at the drill manufacturer's recommended speeds for soft steel. In Fig. 20 we have shown the data obtained from makers of drills—from Capital and Novo steels—relatively to the variation of the number of revolutions of the drill with its size, when the rate of feed is taken constant for all sizes, and equal $\frac{1}{60}$ in. per revolution. The upper—thin line—curve gives the speeds for lubricated, the lower—thick line—for dry drilling. If the drill diameter be δ in inches, and N the speed of revolution at which its circumference will travel at 70 ft. a minute, then obviously $\pi\delta N/12 = 70$, or $N\delta = 267$. Similarly, when the speed is to be 60 ft. per minute, $N\delta = 230$. These two expressions will be found to fit

TABLE XXVI

Table of Values of Greatest and Least and of Ratio of Greatest to Least Spindle Speeds

	Height of centres, in inches.															
	6 in.	9 in.	12 in.	15 in.	18 in.	21 in.	24 in.	27 in.	30 in.	36 in.	42 in.	48 in.	54 in.	60 in.	66 in.	72 in.
$N_g = \frac{3600}{h}$	600	400	300	240	200	171·4	150	133·3	120	100	85·7	75	66·7	60	54·6	50
$N_g = \frac{2400}{h}$	400	266·6	200	160	133·3	114·3	100	89	80	66·66	57·14	50	44·44	40	36·36	33·33
$N_l = \frac{12,800}{h^3} + \frac{30}{h}$	64·25	20·89	9·9	5·8	3·87	2·81	2·18	1·76	1·475	1·108	0·888	0·741	0·636	0·559	0·499	0·451
$\frac{N_g}{N_l} = \frac{3600h^2}{12,800 + 30h^2}$	9·34	19·15	30·3	41·38	51·68	61	68·8	75·8	81·4	90·2	96·6	101	105	107·3	109·4	110·8
$\frac{N_g}{N_l} = \frac{2400h^2}{12,800 + 30h^2}$	6·23	12·75	20·2	27·6	34·44	40·7	45·9	50·57	54·24	60·16	64·35	67·47	69·9	71·55	72·9	73·9

the two curves of Fig. 20. Using the latter value—for dry drilling—we have, from $\frac{\pi h}{12 \times 16} N_s = 60$

$$\therefore \quad N_s = \frac{3670}{h}, \text{ or, say, } \frac{3600}{h} \text{ for \textit{high-speed lathes}}$$

According to this rule the curve given for N_s in Fig. 19 should not consist of two branches, but should be continuous for all sizes. It is true that, in that case, there is a less satisfactory agreement with practice. The authors can only suggest that the spots plotted for the larger sizes in Fig. 19 more properly represent what they have called (see below) "compromise" lathes; having been designed and executed during the recent transition period between the use of carbon and of high-speed steel.

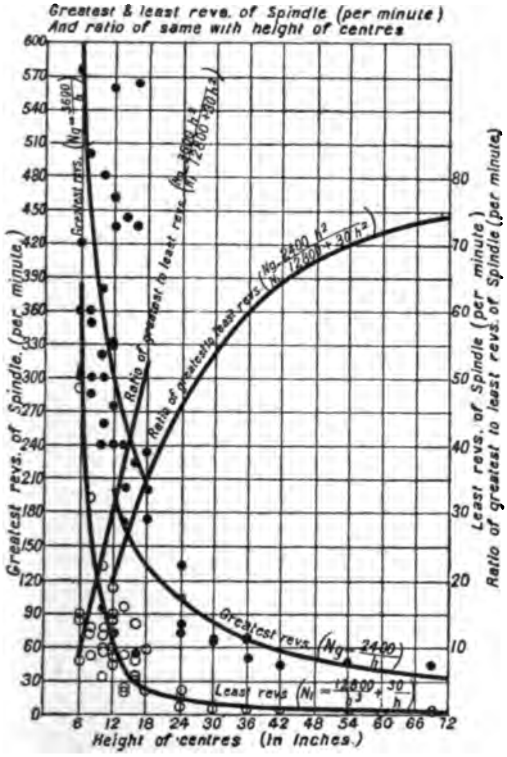


FIG. 19.

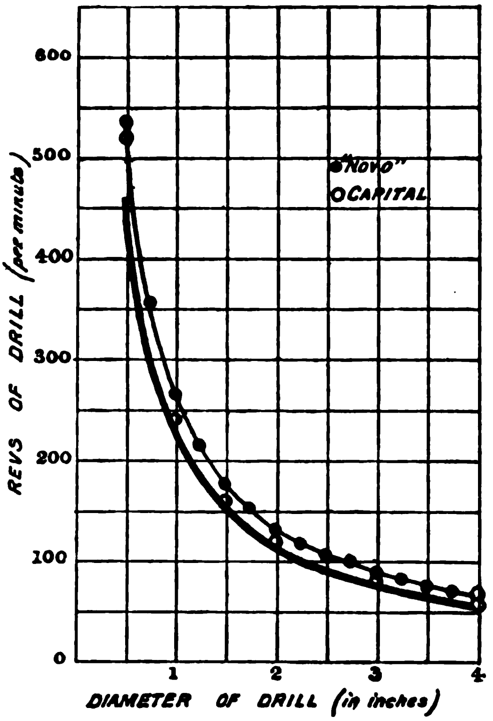


FIG. 20.

CHAPTER VIII

THE NUMBER OF SPEEDS AND THE GEOMETRIC RATIO

THE next step is to find the relation between the range of spindle speeds, and the number of changes of speed to be arranged for within the prescribed limits.

In quite small lathes, where only a few different speeds are required, all that is needed is a couple of duplicate stepped cones, on which the driving belt is shifted. Next in size we have back-geared lathes, where, in addition to the stepped cone driving the spindle direct, it can be coupled to it through two pairs of gears, so that the lathe may be run either by the "single" or the "double" gear. The number of speeds so obtained may also be doubled by having two driving belts from the main to the countershaft, by which means two different countershaft speeds are obtained.

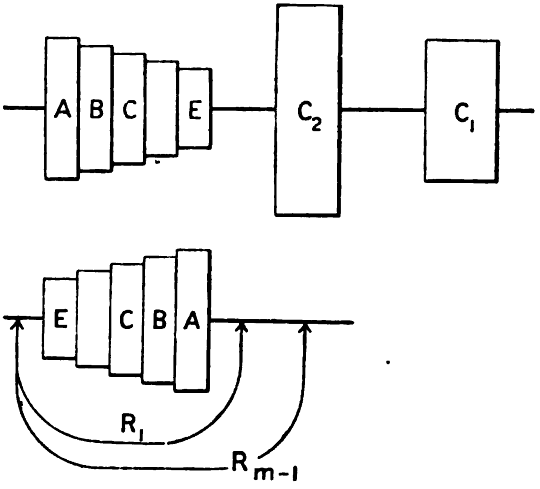


FIG. 21.

The consecutive speeds of the spindle may then be obtained, according to the arrangement adopted, in two different ways.

(A) The belt from the line shaft being on the smaller countershaft pulley, a series of successive speeds is obtained by shifting the belt up the lathe cone from the smallest to the largest step, after which the next lower spindle speed is obtained by moving the cone belt back to its smallest pulley, and putting the slower speed countershaft belt into gear. The belt is then advanced up the cone steps, and so on, through all the possible changes.

(B) The countershaft belt being on the smaller pulley as above, and the lathe cone belt on its smallest step, the spindle will be running at its highest possible speed. To obtain the next lower speed, the cone belt is left unaltered, and the countershaft speed is diminished by changing the drive from the faster to the lower speed. For the next speed the countershaft belt is moved back again, and the cone belt is shifted to the second step; and so on throughout the range.

From the point of view of the cone belt, it is clearly bad practice to adopt method (B) to obtain changes of speed, for there will be a great difference between successive cone-step diameters, making it difficult to shift the belt, as well as giving a bad drive on the lowest cone-step, owing to its small diameter. In what follows, therefore, we shall, as a rule, only refer to method (A):—

Let N_s = highest spindle speed.
N_l = lowest „ „
c = countershaft speed if only one speed.
c_1 = higher „ „ if there are two speeds.
c_2 = lower „ „ „ „
p = number of countershaft speeds.
q = „ steps on the cone.
m = „ different ways cone and spindle can be coupled.
Then S = „ different possible spindle speeds = pqm.
Also let A = diameter of largest cone-step on both lathe and countershaft.
B = „ of second „ „ } Fig. 21.
and so on.
E = „ of smallest „ „
And let r = constant ratio of successive spindle speeds.

Then the table on page 40 shows the relation between these quantities, and deduce the values of the various gear ratios.

Explanation of Table XXVII.—N_1 means the number of revolutions of the spindle when running in single gear (*i.e.* cone coupled directly to the spindle), and with belt on the smallest step of the headstock cone. This is obviously equal to N_s.

When the belt is shifted on to the next larger lathe-cone step without any other change, the spindle speed N_2 is such that $N_1 = rN_2$; and, in the same way, when the belt is on the largest lathe-cone step, the spindle speed N_q will be known from $N_1 = r^{q-1}N_q$, if there are q cone-steps.

From this we can at once deduce the value of $\frac{A}{E}$, the ratio of the diameters of the largest and smallest steps, for we have (Fig. 21)

$$\frac{A}{E} = r\frac{B}{D} = r^2\frac{C}{C} = \ldots = r^{q-1}\frac{E}{A}$$

$$\therefore \frac{A}{E} = r^{\frac{q-1}{2}} = \sqrt{r^{q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

The next lower speed of the spindle N_{q+1} is obtained by shifting the belt back to the smallest lathe-cone step and putting in the first or lowest back gear. This gear must have such a ratio R_1 that $N_1 = r^qN_{q+1}$. From this we at once deduce that $R_1 = r^q$.

Proceeding in the same way, we find that the second back gear, whether it be double or treble, must have a ratio $R_2 = r^{2q}$, and that the highest or $(m - 1)$th gear (where there are m different ways of coupling the cone and spindle together) must have the value

$$R_{m-1} = r^{(m-1)q}$$

TABLE XXVII

One Countershaft Speed

Number of ways cone can be coupled to spindle and name of gear.		Speed number.	Belt on cone step.	Spindle revolutions in terms of highest spindle speed.			Gear ratios.
I	Single gear.	1	E	N_1	$= c\frac{A}{E}$	$= N_g$	$c =$ revolutions of countershaft.
”	”	2	D	N_2	$= c\frac{B}{D}$	$= \frac{N_g}{r}$	
”	”	3	C	N_3	$= c\frac{C}{C}$	$= \frac{N_g}{r^2}$	
”	”	—	—		—		
”	”	q	A	N_q	$= c\frac{E}{A}$	$= \frac{N_g}{r^{q-1}}$	$\frac{A}{E} = r^{\frac{q-1}{2}} =$ ratio $\frac{\text{largest}}{\text{smallest}}$ cone step.
II	First compound gear.	$q+1$	E	N_{q+1}	$= c\frac{A}{ER_1}$	$= \frac{N_g}{r^q}$	$R_1 \quad = r^q \quad =$ first gear ratio.
”	”	$q+2$	D	N_{q+2}	$= c\frac{B}{DR_1}$	$= \frac{N_g}{r^{q+1}}$	
”	”	$q+3$	C	N_{q+3}	$= c\frac{C}{CR_1}$	$= \frac{N_g}{r^{q+2}}$	
”	”	—	—		—		
”	”	$2q$	A	N_{2q}	$= c\frac{E}{AR_1}$	$= \frac{N_g}{r^{2q-1}}$	

III	Second compound gear.	$2q+1$	E	$N_{2q+1} = c\frac{A}{ER_2} = \frac{N_g}{r^{2q}}$	$R_2 = r^{2q}$ = second gear ratio.
,,	,,	$2q+2$	D	$N_{2q+2} = c\frac{B}{DR_2} = \frac{N_g}{r^{2q+1}}$	
,,	,,	$2q+3$	C	$N_{2q+3} = c\frac{C}{CR_2} = \frac{N_g}{r^{2q+2}}$	
,,	,,	—	—	—	
,,	,,	$3q$	A	$N_{3q} = c\frac{E}{AR_2} = \frac{N_g}{r^{3q-1}}$	
				And so on.	
m	$(m-1)$th compound gear.	$(m-1)q+1$	E	$N_{(m-1)q+1} = c\frac{A}{ER_{m-1}} = \frac{N_g}{r^{(m-1)q}}$	$R_{m-1} = r^{(m-1)q}$
,,	,,	$(m-1)q+2$	D	$N_{(m-1)q+2} = c\frac{B}{DR_{m-1}} = \frac{N_g}{r^{(m-1)q+1}}$	= $(m-1)$th gear ratio.
,,	,,	$(m-1)q+3$	C	$N_{(m-1)q+3} = c\frac{C}{CR_{m-1}} = \frac{N_g}{r^{(m-1)q+2}}$	
,,	,,	—	—	—	
,,	,,	mq	A	$N_{mq} = c\frac{E}{AR_{m-1}} = \frac{N_g}{r^{mq-1}} = N_l$	$S_r = \frac{N_g}{N_l} = r^{mq-1} = r^{s-1}$

TABLE XXVIII

Two Countershaft Speeds

Number of ways cone can be coupled to spindle.	Name of gear and countershaft speed.	Speed number.	Belt on cone step.	Spindle revolutions in terms of highest spindle speeds (N_g).			Gear ratios.
I	Single gear. Fast.	1	E	N_1	$= c_1\frac{A}{E}$	$= N_g$.	c_1 = higher countershaft revolutions.
,,	,,	2	D	N_2	$= c_1\frac{B}{D}$	$= \frac{N_g}{r}$	
,,	,,	—	—		—		
,,	,,	q	A	N_q	$= c_1\frac{E}{A}$	$= \frac{N_g}{r^{q-1}}$	$\frac{A}{E} = r^{\frac{q-1}{2}}$.
I	Single gear. Slow.	$q+1$	E	N_{q+1}	$= c_2\frac{A}{E}$	$= \frac{N_g}{r^q}$	c_2 = lower countershaft revolutions.
,,	,,	$q+2$	D	N_{q+2}	$= c_2\frac{B}{D}$	$= \frac{N_g}{r^{q+1}}$	
,,	,,	—	—		—		R_c = ratio $\frac{\text{fast}}{\text{slow}}$ countershaft speed.
,,	,,	$2q$	A	N_{2q}	$= c_2\frac{E}{A}$	$= \frac{N_g}{r^{2q-1}}$	$R_c = \frac{c_1}{c_2} = r^q$.
II	First compound (or back) gear. Fast.	$2q+1$	E	N_{2q+1}	$= c_1\frac{A}{ER_1}$	$= \frac{N_g}{r^{2q}}$	$R_1 = \frac{\text{cone}}{\text{spindle}}$ revolutions with first back gear in.
,,	,,	$2q+2$	D	N_{2q+2}	$= c_1\frac{B}{DR_1}$	$= \frac{N_g}{r^{2q+1}}$	
,,	,,	—	—		—		
,,	,,	$3q$	A	N_{3q}	$= c_1\frac{E}{AR_1}$	$= \frac{N_g}{r^{3q-1}}$	$R_1 = r^{2q}$.

II	First compound gear. Slow.	$3q+1$	E	$N_{3q+1} = c_2\frac{A}{ER_1} = \frac{N_g}{r^{3q}}$	
,,	,,	$3q+2$	D	$N_{3q+2} = c_2\frac{B}{DR_1} = \frac{N_g}{r^{3q+1}}$	
,,	,,	—	—	—	
,,	,,	$4q$	A	$N_{4q} = c_2\frac{E}{AR_1} = \frac{N_g}{r^{4q-1}}$	$R_cR_1 = r^{2q}$.
m	$\left\{\begin{array}{l}(m-1)\text{th compound gear.}\\ \text{Faster countershaft speed.}\end{array}\right\}$	$2(m-1)q+1$	E	$N_{2(m-1)q+1} = c_1\frac{A}{ER_{m-1}} = \frac{N_g}{r^{2(m-1)q}}$	$R_{m-1} = \frac{\text{cone}}{\text{spindle}}$ revolutions with $(m-1)$th back gear in.
,,	,,	$2(m-1)q+2$	D	$N_{2(m-1)q+2} = c_1\frac{B}{DR_{m-1}} = \frac{N_g}{r^{2(m-1)q+1}}$	
,,	,,	—	—	—	
,,	,,	$(2m-1)q$	A	$N_{(2m-1)q} = c_1\frac{E}{AR_{m-1}} = \frac{N_g}{r^{(2m-1)q-1}}$	$R_{m-1} = r^{2(m-1)q}$.
m	$\left\{\begin{array}{l}(m-1)\text{th compound gear.}\\ \text{Slower countershaft speed.}\end{array}\right\}$	$(2m-1)q+1$	E	$N_{(2m-1)q+1} = c_2\frac{A}{ER_{m-1}} = \frac{N_g}{r^{(2m-1)q}}$	$R_{m-1} = \frac{\text{cone}}{\text{spindle}}$ revolutions with $(m-1)$th back gear in.
,,	,,	$(2m-1)q+2$	D	$N_{(2m-1)q+2} = c_2\frac{B}{DR_{m-1}} = \frac{N_g}{r^{(2m-1)q+1}}$	
,,	,,	—	—	—	
,,	,,	$2mq$	A	$N_{2mq} = c_2\frac{E}{AR_{m-1}} = \frac{N_g}{r^{2mq-1}} = N_l$	$R_{m-1} = r^{(m-1)q \times 2}$ for two countershaft speeds.
And, *in general*, for m ways of coupling cone and spindle, q cone steps, and p countershaft speeds, we have		pmq	—	$S_r = \frac{N_g}{N_l} = r^{pmq-1} = r^{s-1}$	$R_{m-1} = r^{(m-1)pq}$.

Also the lowest spindle speed $N_l = N_{mq} = \frac{N_g}{r^{mq-1}}$, so that the ratio of highest to lowest spindle speeds is

$$Sr = \frac{N_g}{N_l} = r^{mq-1} = r^{s-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The following numerical example will show the use of the table :—

Take a 10-in. lathe with 2 double back gears, 1 countershaft speed, and 4 cone steps. Hence $p = 1$, $q = 4$, and $m = 3$.

$$N_g = \frac{3600}{h} = 360$$

$$N_l = \frac{1}{h}\left(\frac{12{,}800}{h^2} + 30\right) = 15, \text{ say}$$

$$S_r = \tfrac{360}{15} = 24$$

Now s, the number of spindle speeds, is

$$s = pqm = 12 \text{ speeds}$$

But $\quad S_r = r^{s-1}$

Therefore $\quad 24 = r^{11}$

or $\quad 11 \log r = \log 24$

or $\quad \log r = \frac{\log 24}{11}$

$$\log r = \frac{1{\cdot}3802}{11} = 0{\cdot}1255$$

$$r = 1{\cdot}335$$

Hence $\quad \frac{A}{E} = r^{\frac{q-1}{2}} = r^{\frac{3}{2}}$

or $\quad \log \frac{A}{E} = \frac{3}{2} \log r = \frac{3}{2} \times 0{\cdot}1255 = 0{\cdot}1883$

and $\quad \frac{A}{E} = 1{\cdot}543$

If $A = 7\sqrt{h} = 22''$, $E = \frac{22}{1{\cdot}543} = 14\frac{1}{4}''$, [$B = 19{\cdot}4''$, and $C = 16{\cdot}8''$] the first back-gear ratio

$$R_1 = r^q = 1{\cdot}335^4$$

or $\quad \log R_1 = 4 \log r = 0{\cdot}5020 = \log 3{\cdot}177$, or $R_1 = 3{\cdot}177$.

Similarly, the second back-gear ratio is

$$R_2 = r^{2q}$$

or $\quad R_2 = r^8 \times \log R_2 = 8 \log r = 8 \times 0{\cdot}1255 = 1{\cdot}0040 = \log 10{\cdot}09$.

In fact, $R_2 = R_1^2 = 10{\cdot}09$.

As a similar numerical example illustrating the use of Table XXVIII we may take the following :—

A 54-in. centres lathe, with the cone off the spindle, two countershaft speeds, three back gears, and four pulleys on the cones. Here $p = 2$, $m = 4$, $q = 4$, so that the number of speeds is $s = 2 \times 4 \times 4 = 32$; and $h = 54$ in.

Then by formula (31)

$$N_g = \frac{2400}{h} = 44 \cdot 5$$

and
$$N_l = \frac{12{,}800}{h^3} + \frac{30}{h} = 0 \cdot 636 \text{ revs. per minute}$$

$$\therefore \frac{N_g}{N_l} = \frac{44 \cdot 5}{0 \cdot 636} = 70$$

Now
$$r = \left(\frac{N_g}{N_l}\right)^{\frac{1}{z-1}}$$

$$\therefore r = 70^{\frac{1}{31}} = 1 \cdot 147$$

Taking $a = 7\sqrt{h} = 51 \cdot 4$ in.—although it might be larger—we can find E, for

$$E = \frac{A}{r^{\frac{q-1}{2}}} = \frac{51 \cdot 4}{1 \cdot 147^{\frac{3}{2}}} = \frac{51 \cdot 4}{1 \cdot 23} = 41 \cdot 8 \text{ in.}$$

and the other two cone pulley diameters are easily found.

Again, the ratio of countershaft speeds is

$$R_c = \frac{c_1}{c_2} = r^q = 1 \cdot 147^4 = 1 \cdot 735$$

The first back-gear ratio is obtained from the general formula

$$R_{m-1} = r^{p(m-1)q}$$

by putting $p = 2$ and $m = 2$; then $R_1 = r^{2q}$; as is also seen directly from the table

$$R_1 = r^{2q} = r^8 = 1 \cdot 147^8 = 3 \cdot 009$$

The second back-gear ratio R_2 in the same way (since m now $= 3$) is got from

$$R_{m-1} = r^{p(m-1)q} = r^{2 \times 2q} = r^{4q} = r^{16} = 1 \cdot 147^{16} = 9 \cdot 05$$
$$\text{or } R_2 = 9 \cdot 05$$

and, finally, the third back-gear ratio R_3 is

$$R_3 = r^{6q} = r^{24} = 1 \cdot 147^{24} = 27 \cdot 25.$$

CHAPTER IX

LATHES FOR HIGH-SPEED CUTTING—THE BELT—EFFECT OF NUMBER OF CONE STEPS, BACK GEARS, AND COUNTERSHAFT SPEEDS UPON ITS WIDTH

In this chapter we shall give our attention to the question of the belt-drive. This is the more important, as the whole design of the headstock is governed by it, and the consideration of the effect upon the size of the belt of obtaining a given number of changes of speed by varying the number of the cone pulleys, of the back gears, or of the countershaft speeds is fundamental to any further advance in the study of rational methods.

The ordinary belt-drives now supplied to lathe headstocks can only be characterized as of very inferior efficiency. The hitherto almost universal custom of mounting the cone upon the spindle and coupling it directly to the latter for the higher spindle speeds is the stumbling-block which proves disastrous to the proper power supply of such lathes.

Owing to this, the belt speed, even when at its highest, is far below what it might and ought to be; and the result is that, when transmitting the power the lathe really requires, the belts are strained much beyond their proper working load, are continually stretching and breaking, and causing delay for their repair.

The charges upon an establishment due to the time the machinery stands idle for belt tightening, repairs, and renewals are out of all proportion greater than what would be incurred by the extra first cost of properly designed headstocks and belts of suitable proportions.

As the purchaser is unaware of these facts, and the maker does not generally supply a more costly article than he is asked for, it seems worth while to give the subject a considerable amount of attention with a view to improvement.

The whole matter turns upon the proper value to be allowed for the working tension per inch of width of a belt, as depending on the speed through the air and the diameter of the pulleys over which it runs.

The ordinary formulæ for the effective pull

$$P = T_2 - T_1 = \frac{2(k-1)}{k+1}T_o$$

where

T_2 = tight side tension
T_1 = slack side tension
T_o = initial tension

and for the ratio of the tensions

$$k = \varepsilon^{\mu\theta} \text{ or } \frac{T_2}{T_1}$$

make no reference either to pulley diameter or to speed, but involve only the coefficient of friction μ, and the angle of embrace θ. Its inapplicability to ordinary driving belts is therefore apparent; but it may be useful for academic exercises about ropes wrapped around square posts.

In the year 1892, and again in 1900, Herr Gehrckens, of Hamburg, gave some very valuable data derived from a long experience with the highest class of belt drives, and from experiments he had made specially to test the driving power of belts at very high speeds. The original papers may be consulted in the *Zeitschrift des Vereins Deutscher Ing.*

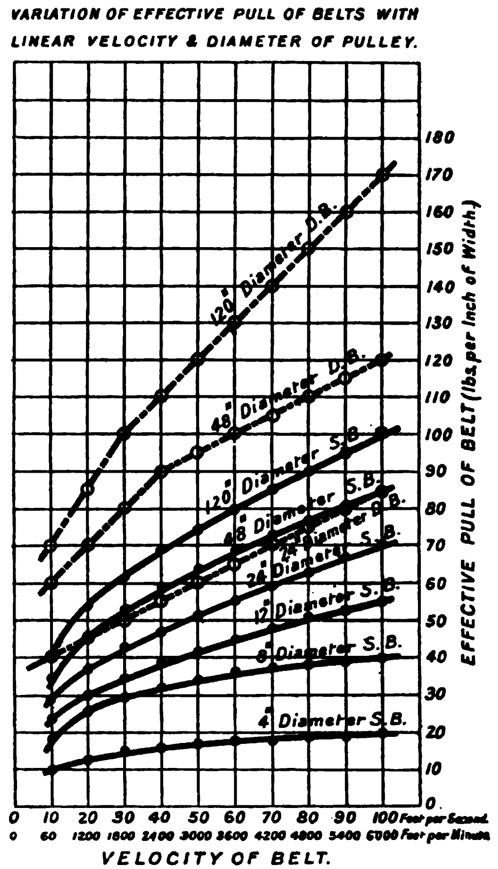


FIG. 22.—VARIATION IN PULL OF BELTS.

The annexed table—XXIX.—gives a digest of his figures in English measure, and the values have been plotted in Fig. 22.

The numbers in the body of the table are the values of the effective tension (P) per inch of belt, whether single or double, which he has found to be allowable for the given speeds (V) in feet per second, and for belts running upon pulleys of the given sizes.

TABLE XXIX

Safe Effective Tension per Inch of Width (Gehrckens)

Single thickness belts.

Diameter of pulley in inches.	Speed in feet per second.									
	10	20	30	40	50	60	70	80	90	100
4	10	13	15	16	17	18	18	19	19	20
8	18	25	29	32	34	36	37	38	39	40
12	23	30	35	39	42	45	48	51	53	55
24	29	37	43	47	51	55	59	63	67	70
48	35	45	53	59	64	69	73	77	81	85
120	40	54	62	69	75	80	85	90	95	100

Double thickness belts.

24	40	45	50	55	60	65	70	75	80	85
48	60	70	80	90	95	100	105	110	115	120
120	70	85	100	110	120	130	140	150	160	170

These figures show that the effective or driving tension of a belt may be increased either by increasing the pulley diameter, or by increasing the speed through the air. In the case of a belt running over a 48-in. pulley, the pull may be raised from 45 lb. to 85 lb. per inch of width, if of single thickness leather, and from 70 lb. to 120 lb. per inch if double, when the speed is increased from 1200 ft. to 6000 ft. per minute. With a speed of 3000 ft. per minute the safe pull is increased from 42 lb. to 64 lb. per inch by increasing the diameter of the pulley from 12 in. to 48 in. diameter.

Let V be the belt speed when it is most heavily loaded. Let P be the allowable effective belt tension per inch of width, as given in Table XXIX. and Fig. 22 from Gehrckens' experience. Then formulæ for these values are as follows:—

$$P = 7\sqrt{D} + \frac{\sqrt{D}}{14}V \text{ – (for single thickness)} \quad . \; . \; . \quad (36)$$

$$P = 7{\cdot}5\sqrt{D} + \frac{\sqrt{D}}{10}V \text{ – (for double thickness)} \quad . \; . \quad (37)$$

Here D = pulley diameter in inches, and V = belt speed in feet per second.

Explanations of Herr Gehrckens' figures, which are of undoubted validity for ordinary workshop practice, have been offered, but there appears to be room for a very useful series of experiments on this subject at the present time.

Assuming for the present that Gehrckens' figures give the most economical results in the long run for a driving belt, when proper regard is had, not merely to first cost of belt, added cost of fast headstock, etc., but also to the diminished annual loss due to stoppages for repairs and to less frequent renewals, consequent upon the low effective tensions which he indicates—especially for slow-speed belts running upon the smaller sizes of pulley—we shall examine the question of the width of belt required for driving lathes of all sizes from 6 in. to 60 in. height of centres.

There will be three cases to consider.

CASE I.—Lathes of small power, where the cone is mounted upon the lathe spindle, and is coupled directly to it for the highest spindle speeds.

CASE II.—Where there is still a cone, but it is mounted upon a separate shaft—occasionally it is mounted on the spindle, but cannot be coupled directly to it—and geared by one or more pairs of wheels to the spindle, even for the highest spindle speeds.

CASE III.—Where there is no cone for varying the speed by shifting the belt; but variation is effected entirely by changes of gearing, and the lathe is driven by a belt running upon a single pulley upon the headstock—fast and loose—and very often straight from the main shafting.

CHAPTER X

THE BELT (*continued*)—CONE MOUNTED UPON SPINDLE

Case I.—**Cone Mounted on Spindle.**

Let GHP = gross horse-power required for taking the standard cut $\left(a = \frac{h^2}{6400}\right)$ at its proper speed $\left(v = \frac{1}{a} + 15\right)$; and on work of any diameter up to $2h$.

Then we have—B being belt width in inches—

$$33{,}000 \text{ GHP} = \text{BP} \times 60\text{V}$$

or

$$\text{B} = \frac{550 \text{ GHP}}{\text{PV}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

If we assume that the belt will be most heavily strained when running at its slowest speed on the largest lathe cone pulley, of diameter A, say, when a cut of standard area is being taken on work of face plate diameter; then the value of V_l, the corresponding belt speed, is obtained as follows :—

n_g = greatest number of cone revolutions.
n_l = least " " "

$$V_l = \frac{\pi A n_l}{60 \times 12}$$

Now we must obtain n_l in terms of N_g, the greatest spindle revolutions, of which we know the value for each size of lathe. But if the cone is directly coupled to the spindle when the latter is doing N_g revolutions, $n_g = N_g$. Suppose we have two countershaft speeds c_1, the faster, and c_2, the slower. Then E being the diameter of the smallest, or qth, cone-step—A that of the largest—on either cone; we have—

$$n_g = c_1\frac{A}{E}, \text{ and } n_l = c_2\frac{E}{A}$$

Therefore

$$\frac{n_l}{n_g} = \frac{c_2}{c_1}\left(\frac{E}{A}\right)^2 = \frac{1}{R}\left(\frac{E}{A}\right)^2 = \frac{1}{r^q} \cdot \frac{1}{r^{q-1}} = \frac{1}{r^{2q-1}} \quad . \quad . \quad . \quad . \quad (39)$$

or

$$n_l = N_g\frac{1}{r^{2q-1}}$$

Thus

$$V_l = \frac{\pi A N_g}{12 \times 60 r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

Now we find that for modern lathes in which the cone is mounted on the spindle, the diameter of the largest cone-step is twice the height of centres—

vide Fig. 23—the sheers being spread, or the bed dropped, so that this is possible. Or, putting $A = 2h$ in the last expression, we have—

$$V_l = \frac{\pi \times 2hN_g}{12 \times 60r^{2q-1}} = \frac{hN_g}{120r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Inserting the proper value of N_g in this expression, which is $N_g = \frac{3600}{h}$ for lathes up to 18-in. centres, we obtain finally—

$$V_l = \frac{h \times 3600}{120 \times hr^{2q-1}} = \frac{30}{r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

The similar expression for the greatest belt speed—when there are two countershaft speeds—is—

$$V_g = \frac{30}{r^{\frac{q-1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

It is clear that, for a given size of lathe—that is, a given value of $\frac{N_g}{N_l}$—V_l is greater the smaller r^{2q-1} is; *i.e.* the smaller r and q are. Now $\frac{N_g}{N_l} = r^{pmq-1} = r^{s-1}$; and if p and m be fixed, s depends upon q only, and r is smaller the greater q is. Thus, since as q increases r diminishes, the resulting effect upon r^{2q-1}, *i.e.* upon V_1, is but small, provided $\frac{N_g}{N_l}$ remains constant.

On the other hand, for a given size of lathe, if p and q be fixed, and the number of speeds varied by altering m—one plus the number of back gears—s increases, of course, proportionately with m, and r suffers diminution. Therefore, r^{2q-1} is diminished with increasing m, and V_l is therefore increased.

(The percentage increase of V_l, *i.e.* the percentage by which the width of the belt B is lessened, due to a given increase of m, is greater the larger the value of $\frac{N_g}{N_l}$, *i.e.* the larger the lathe.)

All this is best seen by working out concrete cases; and for this purpose we have taken the case of a 6-in., a 12-in., and an 18-in. lathe; and for each, determined the values of r, V_l, and B for a two, three, four, and five-stepped cone; and for two and three back gears; p being 2 in all cases, and $\frac{N_g}{N_l}$ as in Table XXVI.

The results are shown in the accompanying Table XXX.

The general conclusions to be drawn from Table XXX. are as follows:—

A 6-in. belt is required to take the standard cut at its proper speed upon a 6-in. lathe when there are two back gears, and a 5-in. when there are three, irrespective of the number of steps upon the cone. The ratio of the greatest to the least number of spindle speeds has been taken as given in Table XXVI., and in agreement with ordinary practice for a general purpose lathe. But it is obvious that in special purpose lathes, where that ratio need not be so large, if the same number of changes of speed be supplied as in Table XXX. the geometric ratio r will be correspondingly less, and V_l, the least belt speed,

TABLE XXX

Tables of Belt Widths for Lathes with Cones of Two, Three, Four, and Five Steps Mounted on the Spindle, Two Countershaft Speeds. Effective Tension per Inch of Width, Double of Gehrckens

CASE 1*a*.—TWO BACK GEARS; $m = 3$.

k	6 in.				12 in.				18 in.			
$\frac{N_g}{N_l}$	9·34				30·3				51·68			
No. of steps q	2	3	4	5	2	3	4	5	2	3	4	5
No. of speeds s	12	18	24	30	12	18	24	30	12	18	24	30
Geometric ratio r	1·226	1·140	1·103	1·080	1·363	1·223	1·160	1·125	1·432	1·261	1·188	1·147
$\log r$	0·0883	0·0571	0·0422	0·0335	0·1347	0·0872	0·0644	0·0511	0·1558	0·1008	0·0747	0·0592
$2q-1$	3	5	7	9	3	5	7	9	3	5	7	9
r^{2q-1}	1·841	1·980	1·974	2·002	2·533	2·729	2·823	2·884	2·934	3·192	3·334	3·410
Least speed of belt V_l	16·3	15·5	15·2	15	11·8	11	10·6	10·4	10·2	9·4	9	8·8
Horse-power CHP	8·4				10·4				13·7			
$B = \frac{550\text{CHP}}{PV_1}$	5⅝ in.	6 in.	6 in.	6⅛ in.	6⅜ in.	7 in.	7¼ in.	7⅜ in.	7⅜ in.	8 in.	8⅛ in.	8½ in.
Taking P =	50				75				100			

CASE 1*b*.—THREE BACK GEARS; $m = 4$.

k	6 in.				12 in.				18 in.			
$\frac{N_g}{N_l}$	9·34				30·3				51·68			
No. of steps q	2	3	4	5	2	3	4	5	2	3	4	5
No. of speeds s	16	24	32	40	16	24	32	40	16	24	32	40
Geometric ratio r	1·161	1·103	1·075	1·059	1·256	1·160	1·116	1·091	1·298	1·188	1·136	1·106
$\log r$	0·0649	0·0423	0·0314	0·0250	0·0989	0·0645	0·0478	0·0380	0·1133	0·0746	0·0554	0·0440
r^{2q-1}	1·565	1·627	1·660	1·679	1·982	2·100	2·158	2·198	2·188	2·360	2·443	2·489
Least belt speed V_l	19·16	18·5	18·1	17·9	15·15	14·3	13·9	13·65	13·7	12·7	12·3	12·05
Horse-power CHP	8·4				10·4				13·7			
Taking P =	50				75				100			
B =	4⅞ in.	5 in.	5⅛ in.	5⅛ in.	5 in.	5⅜ in.	5½ in.	5⅝ in.	5½ in.	5⅞ in.	6⅛ in.	6¼ in.

These belt widths have also been plotted on Fig. 23, and marked *bb*.

correspondingly greater, and as the breadth B varies inversely as V_t, the necessary width of strap for the standard cut can be much reduced.

For a 12-in. lathe a 7-in. belt is necessary with two, and a 5½-in. with three back gears. These are not much wider than for a 6-in. lathe, because although r increases and V_t diminishes, owing to the increase of $\frac{N_g}{N_t}$ (for any given q), the product of V_t and P, the allowable tension, is found to rise at about the same rate as the horse-power, owing to the larger value admissible (after Gehrckens) for P upon a cone step double the former diameter.

Similar remarks apply to the 18-in. size.

Comparisons of these belt widths, which it should be again pointed out are based upon an effective tension double that which Herr Gehrckens recommends—with the spots from practice, plotted in the upper part of Fig. 23 (ring and dot)—show that the belts at present supplied are only capable of from one-half to three-quarters of the power really required in these lathes; if even a moderately efficient use of the new steel is to be made (calculated belt widths are shown by lines *bb*). No doubt they are capable of taking heavy cuts, but certainly not at their most efficient speed.

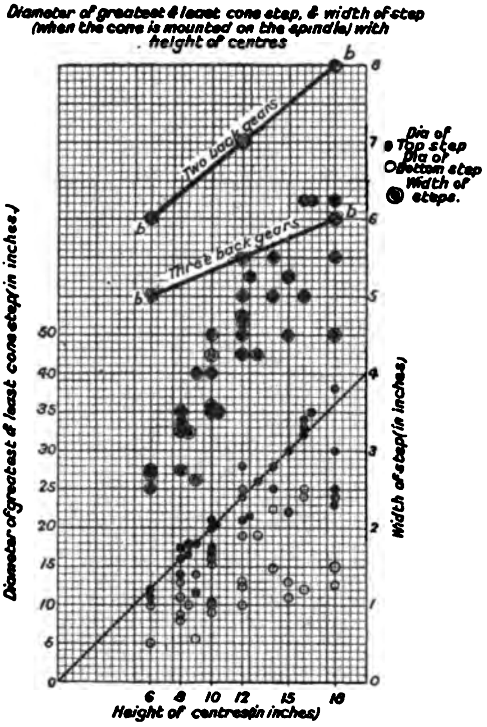


FIG. 23.

Belts so broad as those indicated in Table XXX. are both inconvenient to handle and lead to costly construction, owing to great length of headstock; and the conclusion seems inevitable *that in lathes intended for high-speed steel, which have ordinarily wide ranges of spindle speed, it is bad practice to mount the cone on the spindle, so that the latter can be run single speed.*

CHAPTER XI

THE BELT (*continued*)—CONE MOUNTED ON BACK SHAFT

CASE II.—The cone is mounted on a back shaft, and drives the spindle when running at its highest speed $N_g = \frac{2400}{h}$ through a gear of ratio R_0, which gets greater the larger the lathe. Provisionally we shall lay down the rule that this ratio is to have such a value that the lathe belt speed shall not be greater than 3000 ft. per minute when the belt is running on the largest step of the lathe cone, and the countershaft at its lowest speed. If the largest cone step is again denoted by A, it is found—see Fig. 24—that we may assume the law

$$A = 12\sqrt[3]{h} \quad . \quad (44)$$

for the relation between this diameter and the height of centres. The curve of this expression is shown in upon the figure (24); and we see that practice agrees with, or sometimes exceeds, the values it gives for sizes up to about 40 in., whilst for larger sizes there is not much information available.

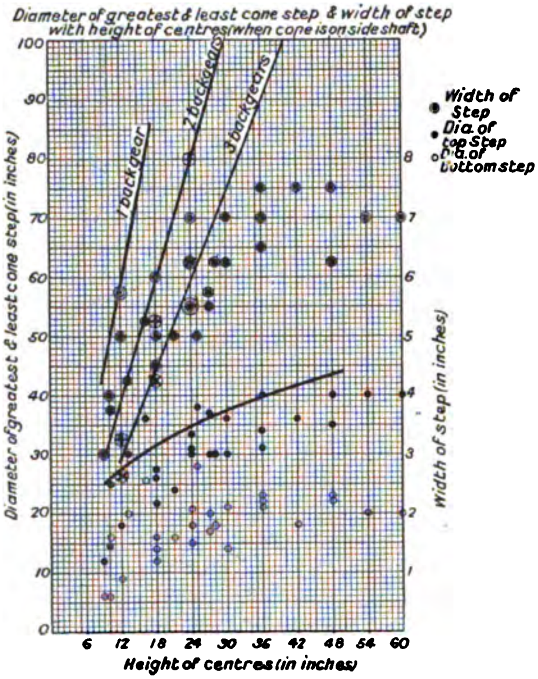


FIG. 24.

If n_l is also, as before, the lowest lathe-cone speed when, *i.e.*, the counter-shaft is running at its slower speed, and the belt is on the largest lathe cone-step; then the slowest belt speed V_l, which is that which governs the design of the belt, as it then has to pull the standard cut on work of face-plate diameter, and at its proper speed for high-speed steel, is got from

$$V_l = \frac{\pi A n_l}{12 \times 60} = \frac{\pi A R_0 N_g}{12 \times 60 \times r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad (45)$$

as before; for $n_g = R_0 N_g$ and $n_l = \frac{n_g}{r^{2q-1}}$.

Now R_o is to be determined by the condition that $V_t = 50$ ft. per sec. So that

$$R_o = \frac{720 r^{2q-1} V_t}{\pi A N_g} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (46)$$

If we take the numbers of speeds, of cone-steps, and of back gears as indicated for the various sizes of lathe in Table XXXI., above, the corresponding geometric ratios for those sizes and conditions will have the values there given; and r^{2q-1} will be known.

Now, putting $V_t = 50$, $A = 12h^{\frac{1}{3}}$, and $N_g = \frac{2400}{h}$, as before, we get—

For a 12-in. lathe $R_o = 4$
„ 18-in. „ $R_o = 6$
„ 24-in. „ $R_o = 6$
„ 36-in. „ $R_o = 9$.

It may be remarked that for all these sizes (and a least belt speed of 50 ft. per second)

$$R_o = \frac{h}{m - 1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (47)$$

or the single-gear ratio is, in this case, equal to the height of centres in inches, divided by the number of back gears.

We may now work out a numerical example under the conditions just specified, and then tabulate the results of a number of other cases obtained in precisely the same way.

To find the width of belt required for an 18-in. lathe for high-speed steel only; the cone being mounted upon a back shaft and driving the spindle when it is running at its highest speed through a 6 to 1 gearing; Gehrckens' figures assumed for the effective belt tension. The number of cone-steps, four; of countershaft speeds, two; the number of back gears, three. Then the geometric ratio is

$$r = \frac{N_g}{N_t}^{\frac{1}{s-1}} = 34^{\frac{1}{13}} = 1{\cdot}121$$

and $r^{2q-1} = 2{\cdot}23.$

Also $A = 12 \times 18^{\frac{1}{3}} = 32.$

Then $V_t = \frac{\pi \times 32 \times 6 \times 2400}{720 \times 2{\cdot}23 \times 18} = 50$ ft. sec.

The effective belt tension is

$$P = \sqrt{A}(7{\cdot}5 + 0{\cdot}1V_t) = 5{\cdot}66 \times 12{\cdot}5 = 70 \text{ lbs. per in.}$$

Therefore $B = \frac{550 \text{ G.H.P.}}{PV_t} = \frac{550 \times 13{\cdot}7}{70 \times 50} = 2{\cdot}46.$

$B = 2\frac{1}{2}$ in., say, *vide* Table XXXI.

Table XXXI. gives the results of such calculations for 12-in., 18-in., 24-in., and 36-in. lathes, taking one, two, three, and four back gears.

If we compare the belt width so obtained with those plotted in Fig. 24,

TABLE XXXI

Table of Belt Widths for Lathes with a Cone of Four Steps Mounted on a Back Shaft, and Geared to the Spindle—at the Highest Speed of the Latter—by One or More Pairs of Wheels of Ratio, $R_o = \frac{h}{m-1}$ = *Height of Centres divided by Number of Back Gears. Two Countershaft Speeds. Effect of Altering the Number of Back Gears upon the Lowest Belt Speed and upon the Size of the Belt for Standard Cut*

h N_g $\frac{N_g}{N_l}$	12 in. 200 20·2				18 in. 134 34·5				24 in. 100 45·9				36 in. 67 60·16				Line No.
No. of back gears . . .	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1
m	2	3	4			3	4			3	4	5		3	4	5	2
No. of speeds s . . =	16	24	32			24	32			24	32	40		24	32	40	3
Geo. ratio r . . . . =	1·222	1·140	1·102			1·168	1·121			1·181	1·132	1·102		1·195	1·141	1·111	4
log r . . . . =	0·0871	0·0568	0·0421			0·0669	0·0497			0·0722	0·0536	0·0426		0·0775	0·0574	0·0456	5
r^{2q-1}	4·97	2·50	1·97			2·94	2·23			3·20	2·37	2·00		3·49	2·52	2·08	6
R_o . . . . =	4 to 1				6 to 1				6 to 1				9 to 1				7
Belt speed V_1	20	40	50			40	50			30	40	50		30	40	50	8
Largest cone step A . =	28 in.				32 in.				35 in.				40 in.				9
P (Gehrckens)	50	58	64			51	71			62	67	72		66	73	79	10
Belt tension per inch of	width taken at 70 lb. throughout				except				. . . .				. .	. .	75	80	11
CHP	10·4				13·7				18·3				31·4				12
Belt width B . . . =	4 in.	2 in.	1⅝ in.			3⅛ in.	2⅛ in.			4⅞ in.	3⅝ in.	3 in.		8¼ in.	5⅝ in.	4⅜ in.	13[1]

[1] Double thickness.

we find them, on the whole, smaller than usual in practice, notwithstanding the fact that the horse-power postulated is in excess of that ordinarily allowed. This is due to the high speed—3000 ft. per minute—we have allowed upon the lathe belt. If we go further into the question we find that such a high belt speed produces countershaft and main shaft speeds of revolution which are altogether too high. Thus we have—

$$\frac{V_g}{V_l} = \frac{r^{2q-1}}{r^{\frac{q-1}{2}}} = r^{\frac{3q-1}{2}} = r^{5\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad (48)$$

Therefore $V_g = 1{\cdot}7\ V_l = 85$ ft. per sec.

if $r = 1{\cdot}102$ (12-in. lathe, three gears).

As $E = \frac{28}{1{\cdot}156} = 24{\cdot}2''$,

therefore $n_g = \frac{85 \times 12}{\pi \times 24{\cdot}2} = 13{\cdot}6$ revs. per sec.

Or the highest cone revs. is 920 per minute.

The highest countershaft revs. is $920\frac{E}{A} = \frac{920}{1{\cdot}150} = 800$ revs. per minute.

Such a speed is only admissible if the counter cone is mounted upon the shaft of an electric motor, and it is because such a method of working, although not as yet generally adopted, appears to offer certain advantages as regards not only first cost, but also running expenses, that the belt widths obtained under the present conditions, and stated in Table XXXI., have been allowed to stand.

It is now proposed to fix R_o in another way. In Table XXXI. the small belt widths obtained are at the expense of a very high speed of countershaft revolution. We may now reverse the process, and fix the allowable lathe belt speed from a consideration of the possible or usual speed of revolution of the countershaft. Assuming a linear speed of the main shaft to countershaft belt of 3000 ft. per minute, the speed of revolution of the latter depends upon the size of the (smaller) pulley diameter allowable. Obviously the larger this is the more economical for the purchaser, as the smaller his belt will be; and since the belt will be replaced many times during the lifetime of the pulley, the less the weight of leather to be renewed the less the maintenance charge.

Taking the smaller countershaft pulley diameter at $7\sqrt{h}$, we obtain the values of the higher countershaft speed, c_1, given in line 9 of Table XXXII. We then obtain R_o as follows:—

$$R_o = \frac{n_n}{N_g} = \frac{\frac{c_1 A}{E}}{N_g} = \frac{c_1 r^{\frac{q-1}{2}}}{N_g}$$

A is maintained as before, equal to $12h^{\frac{1}{4}}$—see curve in Fig. 24. Thus we find—

$$V_l = \frac{\pi A R_o N_g}{720 r^{2q-1}} = \frac{\pi \times 12h^{\frac{1}{4}} \times 50 \times 12 r^{\frac{q-1}{2}}}{720 r^{2q-1} 7\sqrt{h}}$$

$$V_l = \frac{44{\cdot}7}{h^{\frac{1}{4}} r^{\frac{3q-1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

and this is given in line 12 of Table XXXII.

TABLE XXXII

Belt Widths for Lathes with Four Cone Steps. Cone on Back Shaft. Diameter of Smaller Countershaft Pulley = $l\sqrt{h}$.
Belt Speed 3000 *feet per minute*

h N_g N_g N_l	12 in. 200 20·2				18 in. 134 34·5				24 in. 100 45·9				36 in. 67 60·16				Line No.
No. of back gears . . .	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1
m	2	3	4			3	4			3	4	5		3	4	5	2
No. of speeds s . . =	16	24	32			24	32			24	32	40		24	32	40	3
Geo. ratio r . . . . =	1·222	1·140	1·102			1·168	1·121			1·181	1·132	1·102		1·195	1·141	1·111	4
log r . . . . =	0·0871	0·0568	0·0421			0·0669	0·0497			0·0722	0·0536	0·0426		0·0775	0·0574	0·0456	5
r^{2q-1}	4·97	2·50	1·97			2·94	2·23			3·20	2·37	2·00		3·49	2·52	2·08	6
$\frac{A}{E}=r^{\frac{q-1}{2}}$	1·351	1·201	1·156			1·259	1·187			1·283	1·201	1·158		1·306	1·219	1·170	7
R_o . . . . =	3·38	3	2·9			3·75	3·54			4·4	4·15	4		5·6	5·2	5	8
c_1	500	500	500			400	400			344	344	344		286	286	286	9
D . . . . =	24	24	24			30	30			35	35	35		42	42	42	10
A . . . . =	28	28	28			32	32			35	35	35		40	40	40	11
V_l . . . . =	16·7	29·4	36			24	30			21	27	30·5		18·8	24·1	28·2	12
P (Gehrokens)	taken at	60 lb.	through	out													
B . . . . =	5⅝ in.	3¼ in.	2⅜ in.			5¼ in.	4¼ in.			8 in.	6¼ in.	5½ in.		15⅜ in.	12 in.	10¼ in.	18[1]

---

[1] Double thickness.

Gehrckens' constants for the allowable belt tension can now be worked out. They are found to vary but little, and an average value of 60 lb. per inch of width has been taken for all sizes. The formula BPV = 550 GHP now gives B, and this is tabulated as line 13 of Table XXXII.

The values have also been plotted in Fig. 24, and it is not a little surprising that they approximate almost to a linear law passing through zero.

A roughly approximate formula of very great simplicity seems to meet all cases up to 24 in. height of centres, namely—

$$B = \frac{h}{m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

where B is the belt width—double thickness—h is the height of centres, and m is one more than the number of back gears supplied.

For larger lathes the belt widths required for pulling the standard cut on face-plate diameters, at their appropriate speeds, become impossibly great; and this is taken to indicate that *for lathes above* 24 *in. centres the proper mode of driving is by an all-gear head driven through a single pulley at high speed, or by an electric motor.*

As compared with the widths of belt given in Table XXX. for cones mounted directly upon the spindle, Table XXXII. shows the great advantage to be gained by interposing a gear or gears between cone and spindle, even for the highest speeds of the latter, so that the cone can never be coupled directly to it. Even allowing, as was done in Table XXX., double Gehrckens' values of the proper tension, belts of quite impracticable widths are necessary for all but the smallest sizes of lathe in the single speed design if anything like the power is put in which the new steel can make use of.

It would therefore appear that the single-speed design, with cone mounted on spindle, is incapable of giving the results which should be expected by purchasers, except for the smallest sizes of lathe—certainly not larger than 9 in. centres.

We see now, further, that for sizes of lathe above 24-in. centres it will be better to abandon stepped cones altogether, and to secure all variations of speed by means of clutches and gears only, the lathe being driven by a belt upon a single pulley of large diameter running at a high speed. The belt may then run at a speed of 3000 ft. or 4000 ft. per minute, and an effective tension of 100 lb. per inch of width may, according to Gehrckens, be allowed with the best economical results.

This case, known as the "all-gear head," is one which we shall examine in the next chapter.

CHAPTER XII

THE BELT (*continued*)—NO CONE—ALL-GEAR HEADSTOCKS

CASE III.—**All-gear Headstock.**—Belt drives by a single pulley of large size running at high speed.

The problem of this case may be treated more directly than the two first, as no reference need now be made to the geometric ratio or the spindle speed. If f is the cutting stress, a the area of cut, and v the cutting speed, the work spent upon actual cutting is

fav (ft. lbs. per min., if f is in lbs. per sq. in., a is sq. in., and v is ft. per min.).

As a rough approximation—pending further experiments—to the work lost between the main shaft and the tool by reason of journal, gear, and belt friction, we shall assume that when the highest gear is in and the standard cut is being taken upon work of face-plate diameter, the friction horse-power is expressed by the relation

$$\text{FHP} = \frac{h}{6} \quad \dots\dots\dots\dots \quad (51)$$

This means that $33{,}000\,\frac{h}{6} = 5500\,h$ ft. lbs. are spent upon friction per minute.

If P be as before the allowable tension per inch of width B of the belt, and V the speed of the belt in feet per second, the work it is capable of is

$$60\ \text{BPV ft. lbs. per minute.}$$

Hence
$$60\text{BPV} = fav + 5500h$$

or
$$\text{B} = \frac{fav + 5500h}{60\text{PV}} \quad \dots\dots \quad (52)$$

Here
$$f = 2240 \times 115,\ a = \frac{h^2}{6400},\ \text{and}\ v = \frac{1}{a} + 15$$

Therefore
$$\text{B} = \frac{4300 + 10h^2 + 92h}{\text{PV}} \quad \dots\dots \quad (53)$$

If the pulleys are 4 ft. in diameter, so that the belt speed is about 50 ft. per second, with a countershaft speed of 250 revolutions per minute, then $P = \sqrt{D}\,(7{\cdot}5 + 0{\cdot}1V) = 90$ lbs. (by Gehrckens) per inch for a double belt, and formula 53 reduces to

$$B = 0{\cdot}96 + \frac{h}{49} + \frac{h^2}{450}$$

or

$$B = 1 + \frac{h}{50} + \frac{h^2}{450} \sim \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (54)$$

e.g.—

TABLE XXXIII

h = . .	12 in.	18 in.	24 in.	30 in.	36 in.	42 in.	48 in.	60 in.
B = . .	1½ in.	2 in.	2¾ in.	3½ in.	4½ in.	5¾ in.	7 in.	10 in.

These belt widths can only be obtained by giving the belt a speed through the air of 3000 ft. per minute for every size of lathe, and by having the pulleys not less than 4 ft. in diameter.

As it is not usual to supply such large belt pulleys for lathes of less than about 5 ft. centres, the sizes must be redetermined for conditions more in accordance with practice. At the same time there seems no reason why a 4-ft. pulley should not be used for even small lathes, if the design of the headstock were so arranged that the driven fast and loose pulleys might overhang the feed change gear, and be placed at the back of the lathe.

We may assume, for the purposes of our further belt calculations, a countershaft speed of 250 revolutions per minute. There would be no need for a countershaft at all—with the single pulley drive—if the main shop shafting were run at a proper speed. Such a speed as 250 revolutions per minute is, however, quite exceptional, the more usual being 150 to 180 revolutions per minute, or in the more old-fashioned works 120 revolutions per minute. There seems no reason for this but old prejudice and unwillingness to advance with the times. A higher speed of the line shafting would be a great gain all round.

Two cases then fall to be considered. First, when the pulley on the counter—or main—shaft and that on the headstock are both of the same diameter, that diameter being $D = 12\sqrt[3]{h}$. This gives a belt speed, varying with the size of the lathe, from 1800 ft. per minute for a 12-in. lathe to 3000 ft. per minute for a 60-in. Secondly, when the countershaft pulley is kept 4 ft. diameter, and, therefore, the belt speed is 3143 ft. per minute—52·3 ft. per second—for all sizes.

The results of calculating the belt widths based on these two different assumptions are tabulated below—Tables XXXIV. and XXXV.—for sizes of lathe for 12 in. to 72 in.

Lines 7 in each table give the belt widths. Line 9 gives the values of R_o, the ratio of the gear or gears between the driven pulley on the headstock and the spindle when the latter is running at N_g revolutions per minute—its highest speed.

TABLE XXXIV

Table of Belt Widths for Lathes with Single Belt Pulley (all-gear Headstocks); Table A, when both Driving and Driven Pulleys are the same Size, viz. $D = 12\sqrt[3]{h}$ *(vide Line 3), and the Belt runs more slowly for Small than for Large Lathes; Countershaft Speed* 250 *Revolutions per Minute; Gehrckens' Constants.*

												Line No.
h	12	18	24	30	36	42	48	54	60	66	72	1
N_g	200	133	100	80	66·7	57	50	44·44	40	36·36	33·33	2
$12h^{\frac{1}{3}} = D$ . . . . =	27·468	31·44	34·56	37·32	39·6	41·71	43·56	45·36	46·98	48·48	49·92	3
$V = \frac{\pi D}{12} \times \frac{250}{60}$	30	34·3	37·7	40·7	43·2	45·5	47·5	49·5	51·25	52·9	54·5	4
$P = \sqrt{D}(7\cdot5 + 0\cdot1V)$ . . . . =	55	61·3	66·3	70·5	74·4	77·8	80·8	83·8	86·5	89	91·3	5
PV . . . . =	1650	2100	2500	2870	3220	3590	3840	4150	4430	4710	4970	6
$B = \frac{550\,GHP}{PV}$	$3\frac{1}{2}$	$3\frac{5}{8}$	4	$4\frac{5}{8}$	$5\frac{3}{8}$	$6\frac{1}{8}$	$7\frac{1}{8}$	8	$9\frac{1}{8}$	$10\frac{1}{8}$	$11\frac{1}{4}$	7
GHP	10·4	13·7	18·3	24	31·4	40	50	61	74	87	102	8
$R_o = \frac{250}{N_g} = \frac{250h}{2400} = \frac{h}{9\cdot62}$	1·25	1·88	2·5	3·13	3·75	4·4	5·0	5·63	6·25	6·88	7·5	9

TABLE XXXV

Table B, when Driving Pulley is 4 ft. Diameter for all Sizes, but the Driven Pulley is as in Table A ($12h^{\frac{1}{3}}$); *Countershaft Revolutions* 250, *therefore Belt Speed about* 3000 *ft. per Minute throughout, Gehrckens' Constants.*

												Line No.
h		12	18	24	30	36	42	48	54	60	66	72	1
N_g	=	200	133	100	80	66·7	57	50	44·44	40	36·36	33·33	2
$12h^{\frac{1}{3}} = D$	=	27·468	31·44	34·56	37·32	39·6	41·71	43·56	45·36	46·98	48·48	49·92	3
V	=	52·5	52·5	52·5	52·5	52·5	52·5	52·5	52·5	52·5	52·5	52·5	4
$P = \sqrt{D}(7{\cdot}5 + 0{\cdot}1V)$	=	67	71·5	75	77·8	80·3	82·2	84·2	85·7	87·3	88·6	90	5
PV		3520	3750	3940	4090	4210	4310	4420	4490	4580	4650	4720	6
$B = \dfrac{550\,GHP}{PV}$		$1\frac{5}{8}$	2	$2\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{8}$	$5\frac{1}{8}$	$6\frac{1}{4}$	$7\frac{1}{2}$	9	$10\frac{3}{8}$	$11\frac{7}{8}$	7
GHP		10·4	13·7	18·3	24	31·4	40	50	61	74	87	102	8
$R_o = \dfrac{1000}{N_g\,h^{\frac{1}{3}}}$		2·18	2·88	3·47	4·03	4·55	5·06	5·52	5·94	6·4	6·8	7·2	9
Revolutions of driven pulley $= \dfrac{1000}{h^{\frac{1}{3}}}$ . . .		437	383	347	322	303	288	276	264	255	248	240	10

Lines 7 and 9 have been plotted in Fig. 25, as also spots, showing the latest values from practice for this style of design.

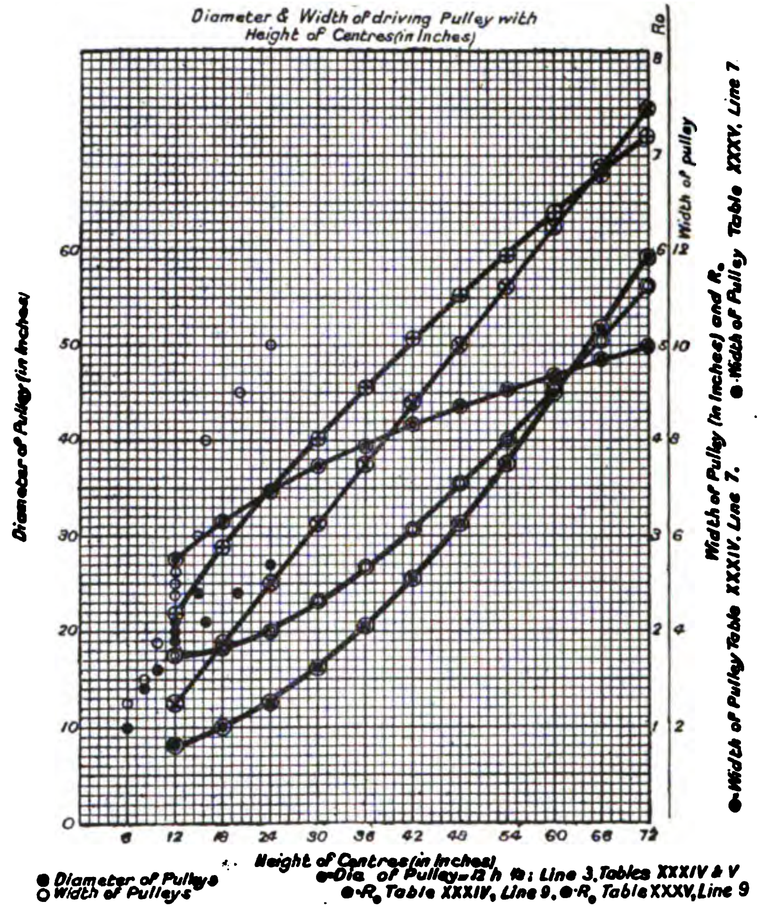


Fig. 25.—All-gear Head—Pulley Diameter and Belt Width.

The improvement to be effected by increased countershaft—or main shaft—speed and larger pulleys is very clearly shown. The driving belts are thereby reduced to a reasonable size, even for the largest lathes.

General Remarks on the Belt Drive.

The power which can be transmitted by a belt B inches wide, travelling with a speed V feet per second, is PBV foot-lbs. per second, or $\frac{PBV}{550}$ H.P., where P is the effective tension per inch of belt. The values of P deduced from Gehrckens' experiments have been given in Chapter IX. They are greater as the pulley diameters are larger and the speed through the air is greater. For the belt speeds usual on a lathe whose cone is mounted on the spindle, this speed is about 15 feet per second, and the effective tension, according to Table XXIX. (p. 48), would be no more than 40 or 50 lb. per inch of widths for a double-thickness belt. The horse-power transmitted per inch of belt at 15 feet per second is then—

$$\frac{\text{PBV}}{550} = \frac{50 \times 1 \times 15}{550} = 1{\cdot}37 \text{ H.P. per inch of width}$$

For a lathe having a cone mounted on a back-shaft, the belt speed may be increased to 50 or 60 feet per second. This means a safe working tension of 60 to 75 lbs. per inch, and a transmitted horse-power 50 per cent. greater than before—say about 2 H.P. per inch of width.

In the case of a lathe having an all-gear headstock, the driving pulley may be of larger diameter than with either of the above designs, in which the belt, when driving the spindle at its highest speed, runs upon the smallest step of the lathe cone. There is also no reason why the speed through the air should not be maintained at 80 or 100 feet per second, if only the line shaft runs at 200 or 300 revolutions per minute (as it ought to do), or an electric motor be used. In that case the allowable tension per inch of width may be safely taken at from 100 to 120 lbs. per inch of width, and the horse-power transmitted will rise to from 2·8 to 4 H.P. per inch of width.

A more particularized inquiry into this question is given in Chapter X. for the first, Chapter XI. for the second, and Chapter XII. for the third of these cases for high-speed steel lathes. Examination will show that the horse-powers per inch of width of belt given in Tables XXX. to XXXV. are on a somewhat more generous scale than the above, this being due to the fact that in Chapters X., XI., and XII. belt tensions double of Gehrckens' best values were allowed.

The question as to which is the better estimate is one which can only be decided by the purchaser's views regarding all-round economy. If he selects narrow belts and high effective belt tensions, he will get a much cheaper lathe; but his bill for repairs, renewals, and stoppages will very soon wipe off the initial saving. By selecting a more expensive lathe, with faster running or broader belts, he will effect an economy in the long run, besides leaving a greater reserve of possible power in the machine for emergencies.

In view of such comparative estimates of the available power and durability of different designs of lathe in respect of their belt drives, it will be well to set forth here in a collected form the formulæ for the various cases. This is the more necessary since we have not, so far, considered the effect upon the power supplied of those designs in which, with two or more countershaft speeds, the consecutive changes of spindle speed are effected by shifting the countershaft belt between every change of cone step. Reference to this question was made in Chapter VIII. (pp. 38 and 39, *q.v.*), but its discussion was postponed as being of less importance than that of the more usual and better method there described as method A.

Employing the symbols specified on p. 39, and the reasoning of p. 50, we easily get the following results for the two ways of working :—

METHOD A.	METHOD B.
$\frac{\text{A}}{\text{E}} = r^{\frac{q-1}{2}}$	$\frac{\text{A}}{\text{E}} = r^{\frac{p(q-1)}{2}}$
$\frac{n_f}{n_l} = r^{pq-1}$; $\frac{c_1}{c_p} = r^{(p-1)q}$	$\frac{n_f}{n_l} = r^{pq-1}$; $\frac{c_1}{c_p} = r^{p-1}$

METHOD A—*continued.*	METHOD B—*continued.*

Now $S_r = \frac{N_g}{N_l} = r^{s-1}$; therefore $r = S_r^{\frac{1}{s-1}}$ for both methods.

METHOD A—*continued.*	METHOD B—*continued.*
But $V_g = \frac{\pi E n_g}{12 \times 60}$	$V_g = \frac{\pi E n_g}{12 \times 60}$
$\therefore V_g = \frac{\pi A N_g}{720 S_r^{\frac{q-1}{2(s-1)}}}$	$V_g = \frac{\pi A N_g}{720 S_r^{\frac{p(q-1)}{2(s-1)}}}$
$V_g = \frac{\pi A N_g}{720 S_r^a}$ $\left[a = \frac{q-1}{2(pqm-1)}\right]$	$V_g = \frac{\pi A N_g}{720 S_r^b}$ $\left[b = \frac{p(q-1)}{2(pqm-1)}\right]$

Hence the greatest horse-power which can be transmitted is—

METHOD A—*continued.*	METHOD B—*continued.*
$HP_g = \frac{PBV_g}{550} = \frac{PB}{550} \frac{\pi A N_g}{720 S_r^a}$	and $HP_g = \frac{PBV_g}{550} = \frac{PB}{550} \frac{\pi A N_g}{720 S_r^b}$
$= \frac{\pi 2h \times 3600}{550 \times 720h} \frac{PB}{S_r^a}$	$= \frac{\pi 2h \times 3600}{550 \times 720h} \frac{PB}{S_r^b}$
$HP_g = \frac{PB}{17{\cdot}5 S_r^a}$	$HP_g = \frac{PB}{17{\cdot}5 S_r^b}$

Similarly, HP_l, the least horse-power, or that which the belt can transmit when running at its slowest speed and mounted on the largest lathe cone step, is—

$$HP_l = \frac{PBV_l}{550} = \frac{PB}{550} \frac{\pi A}{12 \times 60} n_l = \frac{PB\pi A}{550 \times 720} \frac{N_g}{r^{pq-1}}$$

for both methods A and B. Since, as before —

$$r = S_r^{\frac{1}{s-1}}$$

$$\therefore HP_l = \frac{PB\pi A}{550 \times 720} \frac{N_g}{S_r^{\frac{pq-1}{pqm-1}}}$$

$$= \frac{\pi \times 2h \times 3600}{550 \times 720 \times h} \frac{PB}{S_r^c} \qquad \left[c = \frac{pq-1}{pqm-1}\right]$$

$$= \frac{PB}{17{\cdot}5 S_r^c}$$

Thus the horse-power which can be supplied per inch of belt may be taken to be, for lathes in which $A = 2h$ and $N_g = \frac{3600}{h}$,

	With cone on spindle.	With cone off spindle.	With all-gear headstock.
Method A— $HP_g =$	$\frac{2{\cdot}86}{S_r^a}$ $(P = 50)$	$\frac{4{\cdot}8R_o}{S_r^a}$ $(P = 75$ $R =$ single gear ratio)	$5{\cdot}72R_o$ $(P = 100$ $R_o =$ single gear ratio)
Method B— $HP_g =$	$\frac{2{\cdot}3}{S_r^b}$ $(P = 40)$	$\frac{3{\cdot}5R_o}{S_r^b}$ $(P = 60)$	
For both methods— $HP_l =$	$\frac{2{\cdot}3}{S_r^c}$ $(P = 40)$	$\frac{3{\cdot}5R_o}{S_r^c}$ $(P = 60)$	$5{\cdot}72R_o$

CHAPTER XIII

LATHES FOR CARBON STEEL—SPEED, CUT, AND POWER

Lathes for Carbon Steel

It is proper that we should now consider the design of lathes for carbon steel, as the low speeds which are used for finishing, in many cases, render the use of the old brand of steel not only possible, but economical. An obvious suggestion is that a general-purpose lathe should be designed so as to permit of the use not only of high-speed steel for roughing out, but also of common steel for finishing. Before deciding whether any such compromise is possible or desirable, we must make an inquiry into the modifications necessary in the rules above established for high-speed steel lathes, when the use of carbon steel only is contemplated.

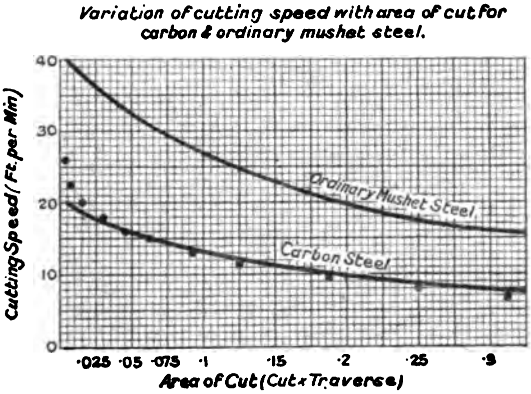


Fig. 26.

The depth of cut, $\frac{h}{40}$, and the width of traverse, $\frac{h}{160}$, will be taken the same as before, so that the expression for the area is

$$a = ct = \frac{h^2}{6400} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

With regard to cutting speed, we have shown, on Fig. 26, a number of values obtained from practice for common, and for ordinary Mushet, steel for the speeds ordinarily allowed with different areas of cut. A simple formula, representing the relation between this speed (v) and the area (a), is

$$v = \frac{4}{a + 0{\cdot}2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

The similar formula for ordinary Mushet is

$$v = \frac{8}{a + 0{\cdot}2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

Thus we obtain for the horse-power required for cutting with carbon steel

$$\text{GHP} = 7{\cdot}8av = 7{\cdot}8\frac{h^2}{6400} \times \frac{4}{\frac{h^2}{6400} + 0{\cdot}2} = \frac{31h^2}{h^2 + 1280} \quad . \quad . \quad (58)$$

The gross horse-power, calculated by this formula, is given in the annexed table for a series of lathes with heights of centres from 6 in. to 6 ft.

Table XXXVI. is, of course, partly a repetition of Table XXV., already given for high-speed lathes, but the lines for cutting speed and horse-power are new. The data are plotted in Fig. 27 as ordinates on a base of height of centres, and may be compared with those given in Fig. 18 for high-speed steel.

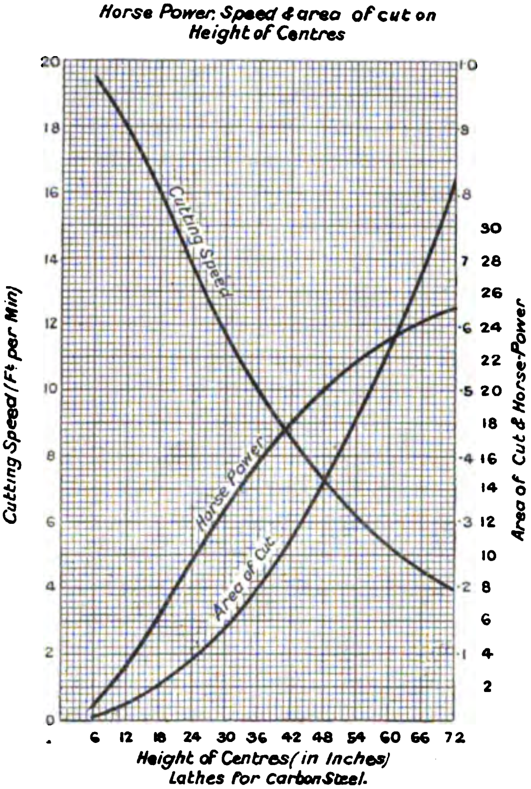


FIG. 27.

SPEEDS AND GEARS.

Using the same notation as that already established on p. 34, *ante*, we have to revise the conditions there postulated in the following manner.

The value of N_g, the highest spindle speed which has been usual in practice, does not appear to have been hitherto fixed by any rational consideration. This is evident from a glance at the full-dot spots plotted in Fig. 28, which are results from recent designs of lathes using carbon steel. With some makers N_g is the same for all sizes of lathe; with others it decreases with height of centres, as it should do; whilst others, again, even *increase* N_g as the height of centres gets greater.

All necessary requirements would appear to be complied with by giving a speed of 30 ft. per minute for the purpose of finishing a bored hole or for the drilling of a hole of a diameter not less than $\frac{1}{16}$th of the height of centres.

If the high spindle speeds sometimes supplied have been furnished with a view of enabling polishing to be done, then it need only be remarked that for the smaller sizes of lathe it is more usual in up-to-date shops to move the

TABLE XXXVI

Table giving Depths of Cut, Widths of Traverse, and Area of Cut proper to Lathes from 6 in. to 6 ft. Height of Centres; also Proper Cutting Speed for those Cuts, and the net Horse-power required.

Height of centres = h . . . . . . =	6	9	12	15	18	21	24	27	30	36	42	48	54	60	66	72
Depth of standard cut = $\frac{h}{40}$	0·15	0·225	0·30	0·375	0·45	0·525	0·60	0·675	0·75	0·90	1·05	1·20	1·35	1·5	1·65	1·80
Width of standard traverse = $\frac{h}{160}$ . .	0·0375	0·05625	0·075	0·0937	0·1125	0·1312	0·15	0·1687	0·1875	0·225	0·2625	0·30	0·34	0·375	0·4125	0·45
Area of standard cut = $\frac{h^2}{6400}$	0·00563	0·01265	0·0225	0·352	0·0506	0·069	0·09	0·114	0·14	0·208	0·276	0·36	0·46	0·56	0·68	0·81
Proper cutting speed = $v = \frac{4}{a+0·2}$.	19·5	18·8	18	17	16	14·9	13·8	12·75	11·75	9·95	8·4	7·15	6·06	5·27	4·55	3·96
Gross horse-power $\} = 7·8av = 7·8\left(\frac{4h^2}{h^2+1280}\right)$	0·855	1·85	3·16	4·66	6·3	8	9·7	11·33	12·86	15·7	18·1	20·1	21·7	23	24·2	25

TABLE XXXVII

Table of Values of Greatest and Least Spindle Revolutions for Lathes using Carbon Steel, enabling a Hole of Diameter $\frac{h}{16}$ to be drilled with High-speed Steel, and giving a Speed of $\frac{4}{(a+0·2)}$ on the Standard at Face-plate Diameter; also the Ratio of these Speeds.

Height of centres h . . . . . =	6	9	12	15	18	21	24	27	30	36	42	48	54	60	66	72
$N_g = \frac{1833}{h}$	305	204	152	122	102	87	76·3	68	61	51	43·6	38·2	34	30·6	27·8	25·4
$N_l = \frac{49,000}{h}\left(\frac{1}{h^2+1280}\right)$ . . .	6·2	4	2·87	2·17	1·7	1·355	1·1	0·904	0·75	0·53	0·383	0·285	0·216	0·1675	0·132	0·105
$\frac{N_g}{N_l} = \frac{3}{80}(h^2+1280) = \frac{h^2}{27}+475 \sim$	49·4	51	53·3	56·5	60	64·5	69·5	75·5	81·5	96·5	114	134·5	157·5	183	211	242
$N_l = \frac{48,900}{(h-\sqrt{h})(h^2+1280)}$ . . =	10·5	6	4·03	2·95	2·22	1·77	1·38	1·11	0·92	0·635	0·455	0·332	0·25	0·192	0·151	0·118
$\frac{N_g}{N_l} = \frac{(h-\sqrt{h})(h^2+1280)}{26·7h}$ . . .	29	34	37·8	41	46	49·2	55	61·2	66·5	80·5	96	105	136	160	184	215

job into a cheaper machine, where a handy labourer can be employed to do the polishing; whilst for larger lathes, where the work is not so easily moved about, the necessary polishing speed of, say, 120 ft. per minute can be obtained on diameters of work even as small as $\frac{h}{4}$, if the above 30 ft. be allowed upon a diameter of $\frac{h}{16}$. (This remark applies also, of course, to small lathes; but in the best shops it is considered better to shift the jobs for polishing purposes into a machine used for that work only.) We then obtain

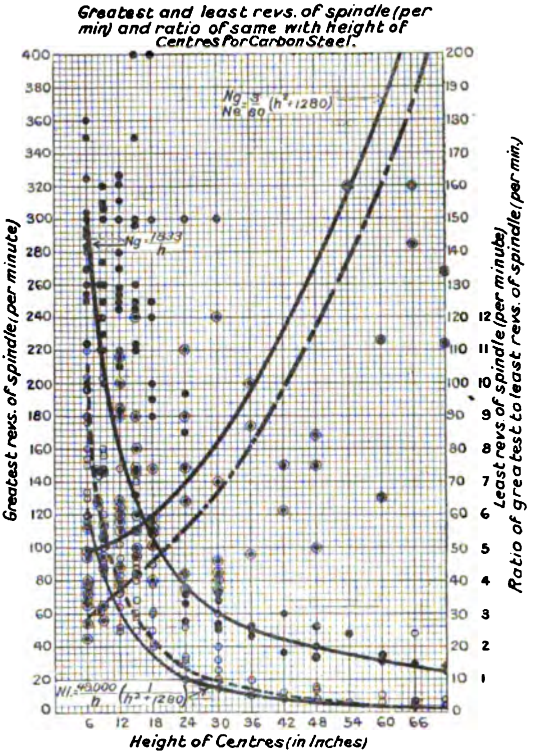


FIG. 28.

$$\frac{\pi h \times N_g}{16 \times 12} = 30$$

or

$$N_g = \frac{16 \times 12 \times 30}{\pi h} = \frac{1833}{h} = \frac{1800}{h} \quad . \quad . \quad . \quad . \quad (59)$$

The smallest number of revolutions of the lathe spindle N_l is determined by the consideration that it must be possible to take the standard cut, $\frac{h^2}{6400}$, upon work of the face-plate diameter and at the appropriate speed, $v = \frac{25,600}{h^2 + 1280}$, found to be usual in practice. We thus get

$$N_l = \frac{12v}{\pi \times 2h} = \frac{12 \times 25,600}{2\pi h(h^2 + 1280)} = \frac{49,000}{h(h^2 + 1280)} \quad . \quad . \quad . \quad (60)$$

Finally, the ratio of the greatest to the least number of spindle revolutions is given by

$$\frac{N_g}{N_l} = \frac{h^2 + 1280}{27} = \frac{h^2}{27} + 475 \quad . \quad . \quad . \quad . \quad . \quad . \qquad (61)$$

These three quantities are set forth numerically in Table XXXVII., and have been plotted as curves on a base of height of centres in Fig. 28. They refer to lathes of from 6 in. to 6 ft. centres. They may be compared with Table XXVI. and Fig. 19 for high-speed steel lathes.

On the same Fig. (28) a large number of values from practice have also been shown. Full black spots are values of N_g; hollow spots, N_l; and spot and ring, $\frac{N_g}{N_l}$. The highly indefinite nature of the data ordinarily employed is very apparent; and it may be said that *hitherto no definite rule has been followed, and no relation has been obtained between the spindle speeds or their ratio and the size of the lathe.* Whether this has been due to the idiosyncrasies of purchasers or the carelessness of makers, it is impossible to say; but it is surely obvious that such a state of confusion ought not to exist.

CHAPTER XIV

LATHES FOR CARBON STEEL—THE BELT—CONE ON SPINDLE

THE BELT DRIVE

IT now becomes necessary to repeat in a modified form for carbon-steel lathes the investigation made in Chapter X. regarding belt width. The same three cases fall to be considered; and we commence at once—referring to p. 50—with:—

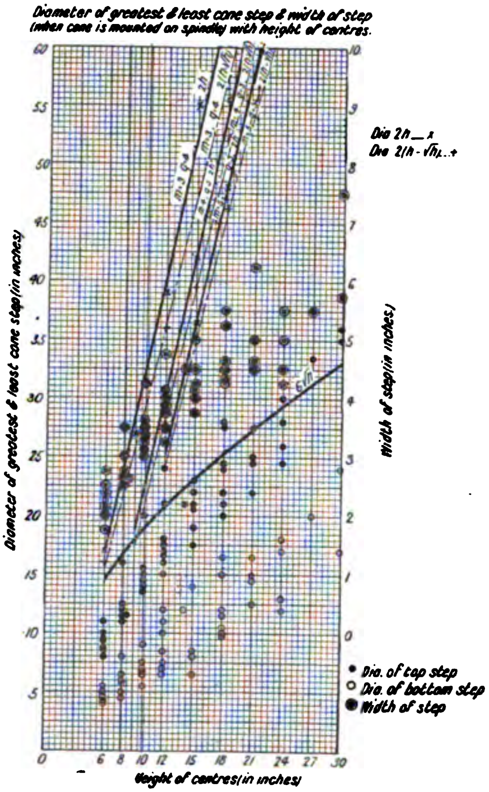


FIG. 29.

CASE I.—*Cone mounted on spindle and connected directly to it for the highest spindle speeds.* The expressions there given for the belt tensions per inch of width 36 and 37; for the gross horse-power, and the belt width in terms thereof (38), remain the same as before. Equations 39 and 40 also hold good if there are two countershaft speeds, and q cone pulleys; but equation 41 will require a slight alteration, as we now propose to assume a value of the diameter of the largest cone step less than $2h$—that assumed on p. 50—in order to be in better agreement with common practice. By reference to Fig. 29, and to the values of the largest cone step diameter there plotted (full dots), it will be seen that a curve having the equation $A = 6\sqrt{h}$ gives a good approximation to average values obtained from recent practice.

Putting this value into expression (40) we have

$$V_1 = \frac{\pi \times 6\sqrt{h}N_q}{12 \times 60r^{2q-1}} = \frac{\pi \times 6\sqrt{h} \times 1833}{720r^{2q-1} \times h} = \frac{48}{\sqrt{h}r^{2q-1}} \quad . \quad . \quad (62)$$

Similarly, the highest cone-belt speed is

$$V_g = \frac{48}{\sqrt{h}\,r^{\frac{q-1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (63)$$

We must now repeat the computation of the items as given in Tables XXX., XXXI., or XXXII.; but for the altered conditions now postulated.

It may here be well to work out one example fully.

EXAMPLE.—*To calculate the width of single belt required upon a 9-in. lathe for carbon steel; cone mounted on spindle; two countershaft speeds; two back gears; three and four cone steps; the allowable tension being taken double of that recommended by Gehrckens;* $N_g = 204$; $N_l = 4$; $\frac{N_g}{N_l} = 51$.

We must first determine the geometric ratio r. It is known from the formula $\frac{N_g}{N_l} = r^{s-1}$, when s is the number of speeds. Now, $s = pqm = 2 \times \begin{pmatrix}3\\ \text{or}\\ 4\end{pmatrix} \times 3 = \begin{pmatrix}81\\ \text{or}\\ 24\end{pmatrix}$, according as three or four cone steps are assumed. So that, from $r = \left(\frac{N_g}{N_l}\right)^{\frac{1}{s-1}}$, we have $\log r = \frac{\log \frac{N_g}{N_l}}{s-1} = \frac{\log 51}{17 \text{ or } 23} = \frac{1{\cdot}7076}{17 \text{ or } 23} = 0{\cdot}1004$ or $0{\cdot}0743$, and $r = 1{\cdot}259$ for three, and $1{\cdot}187$ for four cone steps.

Now, $V_l = \frac{48}{\sqrt{h}\,r^{2q-1}}$, and as $\sqrt{h} = 3$; and $(2q - 1) \log r = \log r^{2q-1}$, or

$$\left.\begin{matrix}5\\ \text{or}\\ 7\end{matrix}\right\} \log r = \begin{Bmatrix}0{\cdot}5020\\ 0{\cdot}5201\end{Bmatrix} = \log \left\{\begin{matrix}3{\cdot}177\\ \text{or}\\ 3{\cdot}312\end{matrix}\right.$$

according as $q = 3$ or 4; we have

$$V_l = \frac{48}{3 \times 3{\cdot}177 \text{ or } 3{\cdot}312} = 5{\cdot}04 \text{ or } 4{\cdot}84 \text{ ft. per second}$$

Then we have, from our formula for Gehrckens' belt tension

$$P = \sqrt{A}\left(7 + \frac{V_l}{14}\right) = \sqrt{6h^{\frac{1}{2}}}\left(7 + \frac{5{\cdot}04}{14}\right) = \sqrt{18} \times 7{\cdot}36 = 31{\cdot}2 \text{ and } 2P = 62$$

Now, the horse-power required to take a cut of $\frac{h}{40}$ with a traverse feed equal to one-fourth of this is, by Table XXXVI., 1·85; and the belt width is obtained at once from the formula

$$B = \frac{550\,GHP}{2PV_l} = \frac{550 \times 1{\cdot}85}{62 \times 5{\cdot}04 \text{ or } 4{\cdot}84} = 3{\cdot}25 \text{ in. or } 3{\cdot}4 \text{ in.}$$

This example shows what a small difference in the width of the belt is produced by altering the number of cone steps. It will be observed in the course of the calculation that, although an increase in the number of cone steps diminishes the value of the geometric ratio—the number of back gears and countershaft speeds remaining unchanged—by increasing the number of speeds, yet the term r^{2q-1} is hardly altered, because q increases whilst r diminishes; V_l is, therefore, not much changed, and as P also remains the same, the belt width is not materially affected.

It is quite different, on the other hand, when the number of back gears (that is, $m - 1$) and m are altered. The number of speeds rises proportionally with the number of back gears, and r is correspondingly diminished. But in the term r^{2q-1}, the power $2q - 1$ remains the same, because q is unaltered, and r^{2q-1} therefore gets less with r. Thus V_t is materially changed by altering m; and the belt width is considerably affected.

This is clearly shown in the tables. Take, for example, the 18-in. lathe in Table XXXVIII. By increasing the number of cone steps from three to four, the belt width is changed only from 9·1 in. to 9·4 in. with three back gears, and from 7·6 in. to 7·5 in. with four back gears; whereas, keeping the number of cone steps the same, say four, and increasing the number of back gears from three to four, the belt width is brought down from 9·4 in. to 7·5 in., or by nearly 2 in. From this we are to infer that, as the size of lathe increases, it is necessary to put in more back gears in order to keep down the width of the belt to a reasonable figure, and this fact becomes very evident from a mere inspection of any of the tables.

TABLE XXXVIII

Belt Widths for Lathes using Carbon Steel, Cone on Spindle. Standard Diameter $(2 = h - \sqrt{h})$. *Only One Countershaft Speed.*

h N_g $\frac{N_g}{N_t}$	6 in. 305 29	9 in. 204 34	12 in. 152 38	15 in. 122 41	18 in. 102 46	21 in. 87 49·2	21 in. 76·3 55	1 2 3
Number of back gears .	2	2	2	3	3	4	4	4
m	3	3	3	4	4	5	5	5
q	3	3	4	4	4	4	4	6
s	9	9	12	16	16	20	20	7
Log r.	0·183	0·1915	0·143	0·1075	0·1108	0·0892	0·0916	8
r	1·524	1·554	1·390	1·280	1·290	1·228	1·235	9
q^{-1}	2	2	3	3	3	3	3	10
r^{q-1}	1·845	2·415	2·685	2·101	2·150	1·852	1·882	11
$V_t = \frac{48}{r^{q-1}\sqrt{h}}$	10·6	6·63	5·15	5·9	5·3	5·66	5·2	12
$P = \sqrt{6\sqrt{h}}\left(7 + \frac{V_t}{14}\right)$.	30	32	—	—	—	—	—	13
$P = \sqrt{6\sqrt{h}}\left(7·5 + \frac{V_t}{10}\right)$	—	—	34	36	38	39	41	14
2 P	60	64	68	72	76	78	82	15
Horse-power, GHP . .	0·855	1·85	3·16	4·66	6·3	8	9·7	16
$B = \frac{550\,GHP}{2PV_t}$	0·74	2·4	5	6	8·6	10	12·5	17

Referring especially to Tables XXXVIII., XXXIX. and XL., which cover the case we are now considering, of *the cone mounted upon the spindle and coupled directly to it for the highest spindle speeds*, we see that for lathes above 18 in. centres the widths of belt required to pull the standard cut, area $= \frac{h^2}{6400}$, at the ordinary speeds for carbon steel of 16 ft. per minute and less, become extravagantly great, and we are driven to the conclusion that *for lathes using carbon steel above* 16-*in. or* 18-*in. centres, the cone must be mounted on a back shaft and geared to the lathe spindle by a pair of gears when the highest spindle speeds are being employed.*

In Table XXXIX. the requirement has been fulfilled that the standard cut shall be taken upon work of face-plate diameter ($2h$). Table XXXIX. has been calculated on the assumption that this cut will never be taken on work of diameter which cannot be swung over the saddle. This diameter is found to be $2\ (h - \sqrt{h})$, the depth of the saddle being about equal to $\sqrt{h}$, and this is the fundamental datum underlying Table XL.

The belt widths obtained by calculations precisely the same as that given above, for these two assumptions, standard diameters $2h$ and $2(h - \sqrt{h})$ respectively, have been plotted as ordinates on a base of height of lathe centres in Fig. 29 (*vide* p. 72).

The resulting curves show an approach to linearity which is little short of surprising, when the complexity of the formula for the belt width in terms of the height of centres is considered.

They are all represented by a formula of the type

$$\mathrm{B} = \mathrm{K}h - \mathrm{L}$$

if the spots representing belt widths for different sizes of lathe *with the same number of back gears* are connected together.

The values of K and L may be most concisely given as follows:—

Linear formulæ for belt widths for common—i.e. *carbon steel—lathes with cone on spindle; standard cut to be taken on diameter* = $2h$.

Number of back gears.	Formula.
2	$\mathrm{B} = 0{\cdot}78h - 3{\cdot}25$ in.
3	$\mathrm{B} = 0{\cdot}80h - 5{\cdot}6$ in.
4	$\mathrm{B} = 0{\cdot}82h - 7{\cdot}2$ in.

If we compare these correctly determined values of the belt widths with the values commonly obtaining in practice, as exhibited by the dot and ring spots on Fig. 29, we see that the usual values are, more or less, of the right order of magnitude for lathes between 8-in. and 15-in. centres.

For smaller lathes the ordinarily allowed widths are *too great*, whilst for those above 15 in. the belts usually supplied are *absurdly too small.* In other words, the ordinary make of lathe larger than of 15 in. height of centres (and over 12-in. centres in which there are less than three back gears) with cone mounted on spindle is not capable of taking the standard cut of depth $h/40$ and traverse $h/160$, at a speed of 16 ft. per minute, on work either of face-plate diameter or even of a diameter which can be swung over the saddle. Belt tensions of 200 lb. per inch and over would be required to enable them to do so.

In some cases which have been examined, where the cone is mounted on the spindle, a high belt speed and a correspondingly narrow belt have been obtained by allowing large values of N_g, the maximum spindle speed, to be the rule. In ordinary course, with the proper or ordinary value of N_l (the lowest spindle speed) this would entail a high value of $\frac{\mathrm{N}_g}{\mathrm{N}_l}$; a correspondingly high value of r (the geometric ratio) for a given number of speeds; and a low value of V_b, that speed of the belt upon which its width depends. In such cases the escape from the dilemma has been by the very simple and obvious trick of omitting a large number of speeds from the continuous

TABLE

Table of Belt Widths (B) for Lathes using Carbon Steel. Cones Mounted Three or Four Steps on the Cone. Two,

h	6		9		12		15		
N_g	305		204		152		122		
$\frac{N_g}{N_l}$	49·4		51		53·3		56·5		
Number of back gears	2	2	2	2	2	3	2	3	4
m	3	3	3	3	3	4	3	4	5
q	3	4	3	4	4	3	4	3	3
s	18	24	18	24	24	24	24	24	30
$\log r = \frac{\log \frac{N_g}{N_l}}{s-1}$	0·0996	0·0736	0·1002	0·0743	0·075	0·075	0·0762	0·0762	0·0605
r	1·258	1·185	1·260	1·187	1·189	1·189	1·192	1·192	1·149
$2q-1$	5	7	5	7	7	5	7	5	5
r^{2q-1}	3·148	3·275	3·177	3·312	3·350	2·371	3·415	2·404	2·006
$V_l = \frac{48}{\sqrt{hr^{2q-1}}}$	6·2	6·0	5·0	4·8	4·1	5·8	3·6	5·1	6·2
$P = \sqrt{6\sqrt{h}\left(7 + \frac{V_l}{14}\right)}$	28·5		31·2		—		—		—
$P = \sqrt{6\sqrt{h}\left(7{\cdot}5 + \frac{V}{10}\right)}$	—		—		37		39		40
2 P	56		62		73		78		
Horse-power GHP	0·855		1·85		3·16		4·66		
$B = \frac{550\,\text{GHP}}{2PV_l}$	1·3	1·4	3·2	3·4	5·8	4·1	9	6·4	5·3
$A = 6\sqrt{h}$	15		18		21		24		

TABLE

Table of Belt Widths for Lathes using Carbon Steel. Cone Mounted on Speeds. Three or Four Cone Steps. Two,

h	6		9		12		'5		
N_g	305		204		152		122		
$\frac{N_g}{N_l}$	29		34		3[illegible]		41		
Number of back gears	2	2	2	2	2	3	2	3	4
m	3	3	3	3	3	4	3	4	5
q	3	4	3	4	4	3	4	3	3
s	18	24	18	24	24	24	24	24	30
$\log r = \frac{\log \frac{N_g}{N_l}}{s-1}$	0·086	0·0635	0·09	0·0666	0·0686	0·686	0·07	0·07	0·0556
r	1·219	1·157	1·23	1·166	1·171	1·171	1·175	1·175	1·137
$2q-1$	5	7	5	7	7	5	7	5	5
r^{2q-1}	2·692	2·783	2·818	2·925	3·021	2·203	3·09	2·239	1·897
$V_l = \frac{48}{\sqrt{hr^{2q-1}}}$	7·3	7·05	5·7	5·48	4·6	6·3	4	5·6	6·5
$P = \sqrt{6\sqrt{h}\left(7 + \frac{V_l}{14}\right)}$	28·5	28·4	31·6	31·3	—	—	—	—	—
$P = \sqrt{6\sqrt{h}\left(7{\cdot}5 + \frac{V_l}{10}\right)}$	—	—	—	—	36·2	37·5	38·2	39	39·2
2P	57	56·8	63·2	62·6	72·4	75	76·4	78	78·4
Horse-power GHP	0·855		1·85		3·16		4·66		
$B = \frac{550\,\text{GHP}}{2PV_l}$	1·1	1·2	2·8	3	5·2	3·7	8·3	5·9	5
$A = 6\sqrt{h}$	15		18		21		24		

XXXIX

on Spindle. Standard Diameter of Job = 2h. Two Countershaft Speeds. Three, or Four Back Gears. A = 6√h.

18				21				24				1
102				87				76·3				2
60				64·5				69·5				3
3	3	4	4	3	3	4	4	3	3	4	4	4
4	4	5	5	4	4	5	5	4	4	5	5	5
3	4	3	4	3	4	3	4	3	4	3	4	6
24	32	30	40	24	32	30	40	24	32	30	40	7
0·0775	0·575	0·0614	0·0456	0·0785	0·0583	0·0624	0·0464	0·0802	0·0595	0·0636	0·0473	8
1·195	1·141	1·152	1·111	1·198	1·144	1·154	1·113	1·203	1·147	1·158	1·115	9
5	7	5	7	5	7	5	7	5	7	5	7	10
2·441	2·526	2·028	2·085	2·469	2·560	2·051	2·113	2·518	2·609	2·08	2·143	11
4·6	4·5	5·6	5·4	4·2	4·1	5·1	5·0	3·9	3·7	4·7	4·6	12
—				—				—				13
41				43				44				14
82				85				88				15
6·3				8				9·7				16
9·1	9·4	7·6	7·5	12·2	12·7	10·2	10·3	15·5	16·2	12·7	13·2	17
26				28				30				18

XL

Spindle. Standard Diameter of Job = 2(h − √h). Two Countershaft Three, or Four Back Gears. A = 6√h.

18				21				24				1
102				87				76·3				2
46				49·2				55				3
3	3	4	4	3	3	4	4	3	3	4	4	4
4	4	5	5	4	4	5	5	4	4	5	5	5
3	4	3	4	3	4	3	4	3	4	3	4	6
24	32	30	40	24	32	30	40	24	32	30	40	7
0·0723	0·0536	0·0574	0·0427	0·0736	0·0546	0·0583	0·0433	0·0757	0·0562	0·06	0·0446	8
1·181	1·132	1·141	1·104	1·185	1·134	1·144	1·105	1·191	1·139	1·148	1·109	9
5	7	5	7	5	7	5	7	5	7	5	7	10
2·299	2·372	1·936	1·99	2·333	2·411	1·956	2·01	2·391	2·474	1·995	2·052	11
5	4·75	5·8	5·7	4·5	4·35	5·4	5·25	4·1	4·0	5	4·9	12
—	—	—	—	—	—	—	—	—	—	—	—	13
40·4	40·2	40·7	40·7	41·7	41·6	42·2	42·2	43·0	43·0	43·5	43·4	14
80·8	80·4	81·4	81·4	83·4	83·2	84·4	84·4	86·0	86·0	87·0	86·8	15
6·3				8				9·7				16
8·4	9	7·25	7·5	11·75	12	9·75	10	15	15·5	12	12	17
26				28				30				18

range; so that there is a huge gap in the speed curve between those spindle speeds for which the cone drives it directly and those which are obtained through the back gears. An actual example is exhibited by Fig. 30, in which the spindle speeds are plotted in the ordinary way for this lathe. It was of 18-in. centres, and had the cone mounted on the spindle. The proper value of N_g by our formula is 102 for this size, but by running the cone at a maximum speed of 330 the belt speed is much increased, and the width, instead of being 9 in., was actually 4½ in. This width enabled the lathe to take our standard cut at its proper speed; but only by leaving the gap in the speeds indicated in the figure.

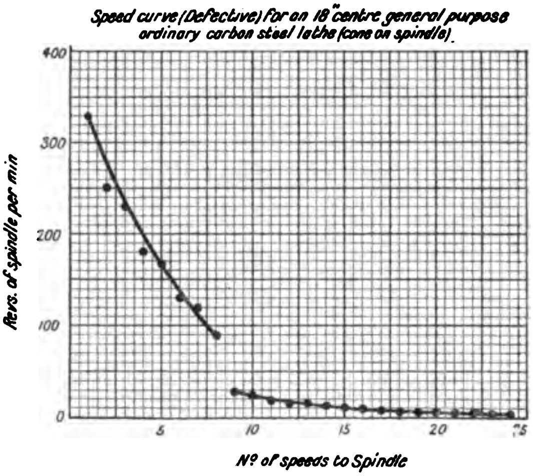


FIG. 30.

The high spindle speeds shown in Fig. 30 serve no useful purpose either for turning, boring, or drilling, and the true speed range of such lathes is comprised within the limits of those for which it is driven by its back gears. As concerns polishing, for which some makers claim the speeds are necessary, we may again point out that the rule we have adopted for N_g gives the respectable polishing speed of 120 ft. per minute on work of any diameter larger than one quarter the height of the centres.

In conclusion, it may be said, with reference to the belt widths shown in Fig. 29, that all the more eminent tool-makers mount their cones on a back shaft for lathes of over 18-in. centres, and the propriety of this procedure is amply confirmed by the width indicated as necessary by the straight line curves of that figure calculated from our rules. We now pass on to the consideration of the next case.

CHAPTER XV

LATHES FOR CARBON STEEL—CONE OFF SPINDLE

CASE II.—*The cone is mounted upon a back shaft and drives the spindle for its highest speeds through a gear of ratio R_o.* Provisionally we shall lay down the rule that $R_o = \sqrt{h}$, which gives ratios which are constructionally convenient. Fig. 31 gives the evidence from practice which justifies the acceptance of this value for the ratio of spur-wheel to pinion diameter through which the higher spindle speeds are obtained. The spindle speeds N_g and N_l being determined in the same way as before, the adoption of this rule for R_o obviously fixes the speed of the cone belt (as also the countershaft revolutions) when running upon any of its steps.

We have, in fact (as on p. 54, *ante*) :—

$$R_o = \frac{n_g}{N_g}$$

Also

$$V_l = \frac{\pi A n_l}{12 \times 60}$$

And

$$\frac{n_l}{n_g} = \frac{1}{r^{2q-1}}$$

$$\therefore V_l = \frac{\pi A n_g}{720 r^{2q-1}} = \frac{\pi A R_o N_g}{720 r^{2q-1}} = \frac{\pi A R_o \times 16 \times 12 \times 30}{720 r^{2q-1} \times \pi h} = \frac{48}{r^{2q-1}} \quad . \quad . \quad (64)$$

From the values of V_l so calculated, and the assumed relation $A = 6\sqrt{h}$—which equally here is seen to be the practical rule for the diameter of the largest cone step (*vide* Fig. 32, full dots and curve)—Gehrckens' constants for the belt tension are obtained, and the belt widths are then easily found.

Throughout these belt calculations (as has been repeatedly pointed out) double the value of Gehrckens' effective tensions have been allowed, as being in better agreement with British practice.

The belt widths have been worked out on the above assumptions for lathes from $h = 12$ in. to $h = 72$ in., and the values have been entered in Tables XLI. and XLII., and plotted in Fig. 32. The full line curves give the belt widths which are capable of pulling the standard cut at its proper speed on a diameter equal to $2h$, the dotted curves on a diameter equal to $2(h - \sqrt{h})$. They are for two, three, and four back gears.

We see that the belt widths are no longer even approximately linear when plotted on the height of centres.

TABLE

Table of Belt Widths for Lathes using Carbon Steel. Cone Mounted on Back
Steps on Cone. One, Two, Three, Four and Five Back Gears. Effect

h	12 in.				18 in.			
N_g	152				102			
$\frac{N_g}{N_l}$	53·3				60			
Number of back gears .	1	2	3	5	1	2	3	5
m	2	3	4	6	2	3	4	6
q	4				4			
$2q-1$	7				7			
s	16	24	32	48	16	24	32	48
Log r	0·115	0·075	0·0557	0·0368	0·1185	0·0773	0·0573	0·0378
r	1·303	1·189	1·137	1·088	1·314	1·195	1·141	1·091
r^{2q-1}	6·383	3·350	2·454	1·809	6·753	3·476	2·519	1·840
$V_l = \frac{48}{r^{2q-1}}$.	7·53	14·3	19·6	26·6	7·10	13·8	19	26·1
$P = \sqrt{6\sqrt{h}}\left(7·5 + \frac{V_l}{10}\right)$.	37·6	40·6	43	46·2	41·4	44·8	47·4	56
$2P$	75	81	86	92	83	90	95	112
Horse-power, GHP . .	3·16				6·3			
$B = \frac{550\,GHP}{PV_l}$	6·2	3	2·08	1·43	11·8	5·6	3·84	2·37
$B = \frac{550\,GHP}{2PV_l}$	3·1	1·5	1·04	0·715	5·9	2·8	1·92	1·185
$R_o = \sqrt{h}$	3·464				4·24			

TABLE

Table of Belt Widths for Lathes using Carbon Steel. Cone Mounted on Back
Four Steps on Cone. One, Two, Three, Four and Five Back Gears. Effect

h	48 in.						
N_g	38·2						
$\frac{N_g}{N_l}$	134·5						
Number of back gears .	1	2	3	4	5	1	2
m	2	3	4	5	6	2	3
q	4						
$2q-1$	7						
s	16	24	32	40	48	16	24
Log r	1·142	0·0925	0·0687	0·0546	0·0452	0·151	0·0985
r	1·1387	1·237	1·171	1·134	1·11	1·416	1·254
r^{2q-1}	9·863	4·44	3·026	2·411	2·072	12·76	4·893
$V_l = \frac{48}{r^{2q-1}}$.	4·87	10·8	15·85	20	23·2	3·76	9·82
$P = \sqrt{6\sqrt{h}}\left(7·5 + \frac{V_l}{10}\right)$.	51·5	55·3	58·5	61·5	63·4	53·7	57·8
$2P$	103·0	110·6	117·0	123	126·8	107·4	115·6
Horse-power, GHP . .	20·1						
$B = \frac{550\,GHP}{PV_l}$	22	9·26	5·97	4½	3·76	31·3	11·3
$B = \frac{550\,GHP}{2PV_l}$	44	18·52	11·94	9	7·52	62·6	22·6
$R_o = \sqrt{h}$	6·9						

XLI

Shaft. $R_o = \sqrt{h}$. *Standard Diameter* $= 2h$. *Two Countershaft Speeds. Four of Increasing Number of Back Gears upon the Belt Width.* $A = 6\sqrt{h}$.

24 in. 76·3 69·5				36 in. 51 58·5					1 2 3
1 2	2 3	3 4	5 6	1 2	2 3	3 4	4 5	5 6	4 5
4 7				4 7					6 7
16	24	32	48	16	24	32	40	48	8
0·1228	0·08	0·0595	0·0392	0·1325	0·0864	0·064	0·051	0·0423	9
1·326	1·202	1·147	1·094	1·357	1·22	1·159	1·125	1·103	10
7·238	3·631	2·61	1·881	8·463	4·025	2·805	2·275	1·977	11
6·63	13·3	18·4	25·5	5·67	11·9	17·1	21	24·3	12
44·2	47·7	50·5	54·5	48·4	52·14	55·26	57·6	59·6	13
88·4	95·4	101·0	109·0	96·8	104·28	110·52	115	119 2	14
9·7				15·7					15
9·14	4·22	2·87	1·92	15·7	6·95	4·56	3½	2·98	16
18·28	8·44	5·74	3·84	31·4	13·90	9·12	7	5·96	17
4·9				6					18

XLI.—*continued.*

Shaft. $R_o = \sqrt{h}$. *Standard Diameter* $= 2h$. *Two Countershaft Speeds. of Increasing Number of Back Gears upon the Belt Width.* $A = 6\sqrt{h}$.

60 in. 30·6 183			72 in. 25·4 242					1 2 3
3 4	4 5	5 6	1 2	2 3	3 4	4 5	5 6	4 5
4 7			4 7					6 7
32	40	48	16	24	32	40	48	8
0·0731	0·058	0·0482	0·159	0·1038	0·077	0·061	0·05075	9
1·183	1·143	1·118	1·442	1·270	1·194	1·151	1·124	10
3·248	2·547	2·175	12·92	5·328	3·459	2·673	2·2526	11
14·75	19	22·1	3·72	9·02	13·9	18	21·3	12
61·2	64	66·2	56·1	60	63·4	66	68·6	13
122·4	128	132·4	112·2	120	126·8	132	137·2	14
23			25					15
7·03	5¼	4·32	33	12·7	7·8	5¾	4·7	16
14·06	10½	8·64	66	25·4	15·6	11½	9·4	17
7·75			8·5					18

TABLE

Table of Belt Widths for Lathes using Carbon Steel. Cone of Four Steps Mounted on a of Wheels of Ratio $R_o = \sqrt{h}$. Standard Diameter = $2(h - \sqrt{h})$. Two Counter- Width of the Belt. $A = 6\sqrt{h}$.

h N_g $\frac{N_g}{N_l}$	12 in. 152 38				18 in. 102 46			
Number of back gears	1	2	3	5	1	2	3	5
m	2	3	4	6	2	3	4	6
q	4				4			
$2q-1$	7				7			
s	16	24	32	48	16	24	32	48
Log r	0·1052	0·0688	0·051	0·0336	0·111	0·0723	0·0537	0·0354
r	1·275	1·171	1·125	1·08	1·291	1·181	1·132	1·085
r^{2q-1}	5·450	3·031	2·275	1·719	5·984	3·207	2·376	1·769
$V_l = \frac{48}{r^{2q-1}}$	8·8	15·81	21·1	27·9	8·2	15	20·2	27·2
$P = \sqrt{6\sqrt{h}}\left(7{\cdot}5 + \frac{V_l}{10}\right)$	38·1	41·3	43·8	46·8	42	45·4	48	56·5
2P	76	82	87	93	84	91	96	113
Horse-power = GHP	3·16				6·3			
$B = \frac{550\,GHP}{2PV_l}$	2⅝	1⅜	1	⅝	5	2½	1¾	1⅓
$B = \frac{550\,GHP}{PV_l}$	5¼	2¾	2	1¼	10	5	3½	2¼
$R_o = \sqrt{h}$	3·464				4·24			

TABLE

Table of Belt Widths for Lathes using Carbon Steel. Cone of Four Steps Mounted a Pair of Wheels of Ratio $R_o = \sqrt{h}$. Standard Diameter = $2(h - \sqrt{h})$. Two Lowest Speed and Width of the Belt. $A = 6\sqrt{h}$.

h N_g $\frac{N_g}{N_l}$	48 in. 38·2 105						
Number of back gears	1	2	3	4	5	1	2
m	2	3	4	5	6	2	3
q	4					4	
$2q-1$	7					7	
s	16	24	32	40	48	16	24
Log r	0·135	0·088	0·0653	0·0518	0·043	0·147	0·096
r	1·365	1·225	1·162	1·127	1·104	1·403	1·247
r^{2q-1}	8·81	4·130	2·865	2·304	2	12·68	4·699
$V_l = \frac{48}{r^{2q-1}}$	5·45	11·62	16·75	20·8	24	3·78	10·25
$P = \sqrt{6\sqrt{h}}\left(7{\cdot}5 + \frac{V_l}{10}\right)$	51·8	55·9	59·1	62	63·8	53·8	58·2
2P	103·6	111·8	118·2	124	127·6	107·6	116·4
Horse-power = GHP	20					23	
$B = \frac{550\,GHP}{2PV_l}$	19¾	8½	5½	4·25	3½	31	10½
$B = \frac{550\,GHP}{PV_l}$	—	17·0	11·2	8·5	7·2	—	21·2
$R_o = \sqrt{h}$	6·9						

XLII

Back Shaft and Geared to the Spindle at the Highest Speed of the latter by a Pair shaft Speeds. Effect of Altering Number of Back Gears upon the Lowest Speed and

24 in. 76·3 55				36 in. 51 80·5					1 2 3
1	2	3	5	1	2	3	4	5	4
2	3	4	6	2	3	4	5	6	5
	4					4			6
	7					7			7
16	24	32	48	16	24	32	40	48	8
0·116	0·0757	0·0562	0·037	0·127	0·083	0·0615	0·049	0·0405	9
1·306	1·191	1·139	1·089	1·340	1·211	1·152	1·119	1·097	10
6·486	3·388	2·474	1·816	7·745	3·811	2·695	2·208	1·921	11
7·41	14·15	19·4	26·4	6·2	12·6	17·8	21·8	25·0	12
44·6	48·3	51	55	48·7	52·6	55·7	58	60	13
89·2	96·6	102	110	97·4	105·2	111·4	116	120	14
	9·7					15·7			15
8	4	2¾	2	14⅜	6½	4⅜	3·4	3	16
16·2	7·8	5·4	3·66	28·6	13·0	8·74	6·8	5·8	17
	4·9					6			18

XLII.—*continued.*

on a Back Shaft and Geared to the Spindle at the Highest Speed of the latter by Countershaft Speeds. Effect of Altering Number of Back Gears upon the

60 in. 30·6 160			72 in. 25·4 215					1 2 3
3	4	5	1	2	3	4	5	4
4	5	6	2	3	4	5	6	5
	4				4			6
	7				7			7
32	40	48	16	24	32	40	48	8
0·714	0·0565	0·047	0·155	0·1015	0·0755	0·06	0·0496	9
1·179	1·139	1·114	1·429	1·263	1·190	1·148	1·121	10
3·161	2·486	2·133	12·83	5·135	3·373	2·63	2·224	11
15·2	19·3	22·5	3·74	9·33	14·2	18·2	21·6	12
61·5	64·5	66·5	56·1	60	63·5	66·5	68·8	13
123·0	129	133·0	112·2	120	127·0	133	137·6	14
	23				25			15
6¾	5·05	4¼	32⅞	12⅜	7⅝	5·7	4⅝	16
13·5	10	8·4	—	24·6	15·30	11·4	9·26	17
7·75					8·5			18

For lathes from about 26 in. to 42 in. height of centres the belt widths allowed in practice agree tolerably well with the values correctly calculated for two back gears.

From 42 in. to 60 in. centres the ordinary belt widths are too small for two back gears; but too great, when compared with our calculated sizes, for three back gears. At the same time it must be remembered that two or even four tools would sometimes be employed with such large lathes, and this may account for the apparent discrepancy. Each tool would, in that case, be taking a good deal less than the standard cut; but it is probable that the belt widths usually supplied are unequal to much more than the equivalent of one tool taking the standard area of cut at the proper speed unless the cutting speed is much reduced for two or more tools.

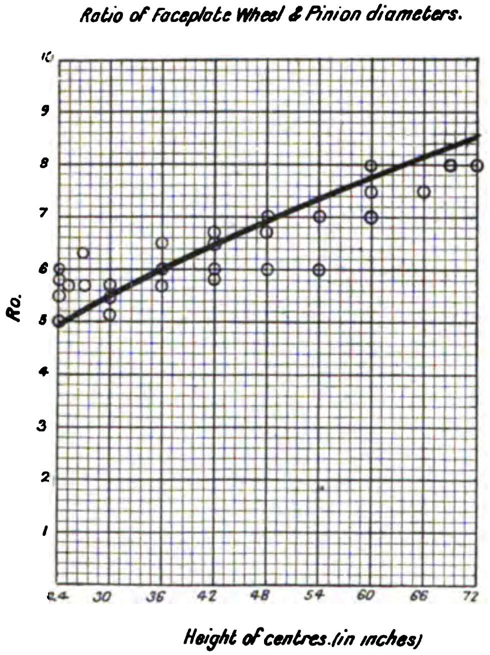


FIG. 31.

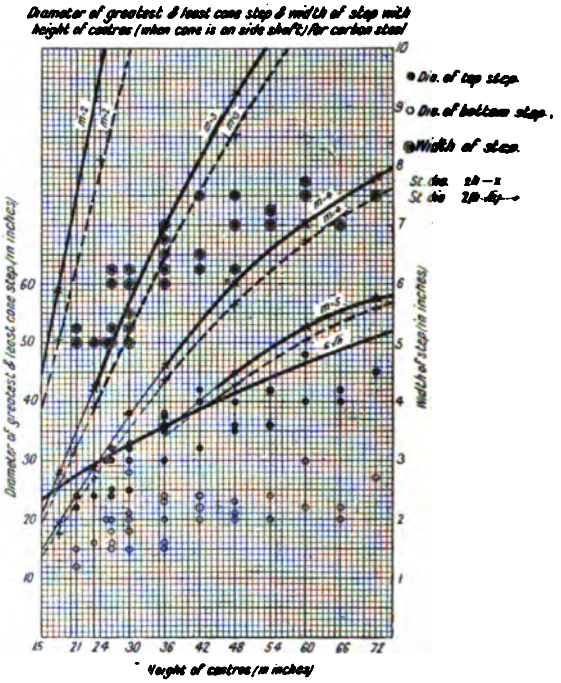


FIG. 32.

The effect of increasing the number of double back gears is very marked. In the case of a 54-in. lathe, a 10-in. belt is required when there are two back gears, a 6½-in. when there are three, and only a 5-in. when there are four. The following simple expressions for the belt widths hold within the specified limits. They are, of course, merely empirical; but may be of use in practice as giving close approximations to the correct sizes. They are of the linear type:—

$$B = Kh \pm L.$$

Approximately correct formulæ for belt widths for lathes using carbon steel; cone on back shaft. Standard diameter = 2h. 4 cone steps.

No. of back gears.	Formulæ.	Correct for lathes.
One	$B = \frac{h}{2{\cdot}5} - 1{\cdot}5$	From 12 in. to 21 in. centres
Two	$B = \frac{h}{5} - \frac{1}{2}$	From 15 in. to 36 in. centres
Three	$B = \frac{h}{10} + 1$	From 36 in. to 66 in. centres
Four	$B = \frac{h}{20} + 2$	From 42 in. to 72 in. centres

In our next chapter the question of a compromise, for general purposes, between a lathe for the use of high speed steel for roughing out and one using carbon steel for finishing, will be considered.

CHAPTER XVI

COMPROMISE LATHE—SPEED—CUT AND POWER

In this chapter we discuss the question of a design of lathe capable of meeting the wants of many purchasers who would not feel justified in ordering a lathe for the use of high-speed steel exclusively; but who would desire to be able to execute finishing cuts with carbon steel at the slower speeds suitable thereto, and so obtain a more perfect finished surface, whilst at the same time the lathe would be supplied with a much greater driving power than would be necessary for carbon steel only, so as to permit of moderately heavy cuts being taken with high-speed steel for roughing out.

Highest Spindle Speed for Compromise Lathe (N_s).—In the case of purely high-speed lathes we gave the formulæ $\frac{3600}{h}$ and $\frac{2400}{h}$ for small and large lathes respectively, as indicating a close approach to the data for N_s obtained from practice. The results of these formulæ were tabulated in Table XXVI., and plotted on Fig. 19 (p. 37, *ante*).

We are now disposed to believe that $\frac{3600}{h}$ *for all sizes* would have been a better rule for high-speed lathes, and that the spots from practice in Fig. 19 for lathes of 18 in. to 72 in. centres more properly represent *compromise lathes*, as having been designed and executed during the recent transition stages of practice from the use of carbon to that of high-speed steel.

At all events, if it be agreed that the high-speed steel lathe is to be capable of drilling at the drill manufacturer's recommended speed of 60 ft. per minute, and that upon holes as small as $h/16$—one-sixteenth of the height of centres—then $3600/h$ becomes the simple and rational rule by which to determine the value of the highest spindle speed for any size of high-speed lathe (*vide* Chap. VII., *ante*).

For *carbon steel lathes* the formula recommended was $N_s = \frac{1800}{h}$, just half of that obtained for high-speed lathes, so that a finish-boring and drilling speed of 30 ft. per minute is obtainable, according to our rule, upon a diameter of $\frac{h}{16}$, and a polishing speed of 120 ft. per minute on a diameter of $\frac{h}{4}$, on our proposed designs of carbon steel lathe.

For the compromise lathe it is suggested that this drilling and finish-boring speed be taken at 45 ft. per minute on a diameter of $\frac{h}{16}$. This would give $N_g = \frac{2750}{h}$ for the compromise lathe (see Table XLV.).

Horse-power and Lowest Spindle Speed for Compromise Lathe (N_l).—For *high-speed lathes* the rule we adopted was that the lathe should be able to run slowly enough to take the "standard cut" for the given size of lathe, *at its proper speed;* that given, viz. by the formula $v = \frac{1}{a} + 15$.

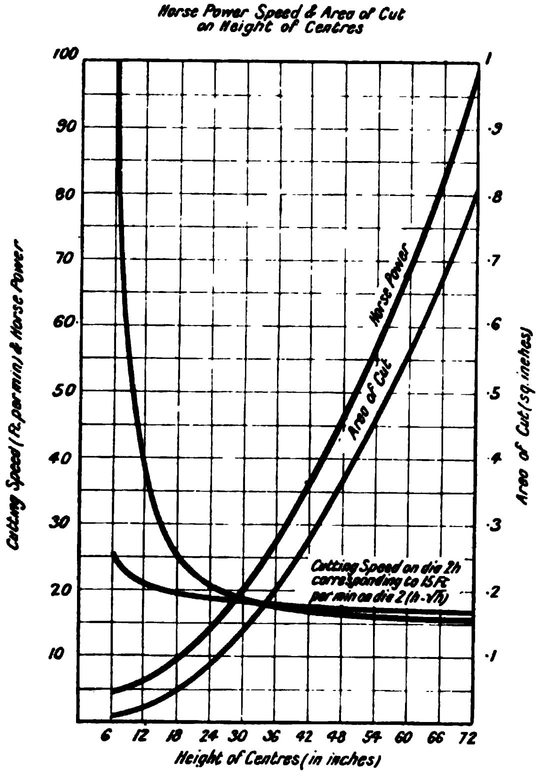


FIG. 33.

For the compromise lathe we propose to adhere to the same standard cut as before, viz. depth, $\frac{h}{40}$; width, $\frac{h}{160}$; area $\frac{h^2}{6400}$.

The horse-power to be supplied is clearly a matter for individual judgment, and for decision in each case upon a consideration of all the circumstances. As a tentative suggestion, we are inclined to offer the opinion that the horse-power to be supplied in the compromise lathe should be determined by allowing that the standard cut above specified is to be taken at the speed given by the formula—

$$v = \frac{1}{2a} + 15 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

The values of this speed for lathes from 6 in. to 6 ft. height of centres are given in Table XLIV.; and the horse-powers required for each size of lathe. They are also plotted in Fig. 33.

In the same figure another speed curve is shown, representing the speed on the face-plate diameter, corresponding to 15 ft. per minute on work of diameter which can be swung over the saddle. One speed curve is that which determines the horse-power, the other is that which governs the slowest spindle speed. The latter speed varies from 25 ft. per minute on small to 17 ft. on large lathes, and this is in accordance with the requirements of practice, inasmuch as the finishing cuts are heavier upon large than upon small sizes of lathe.

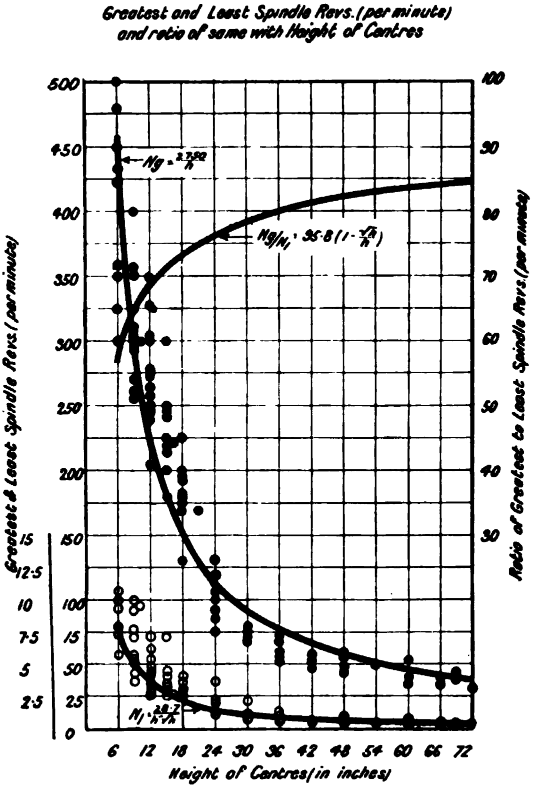


FIG. 34.

For comparison we give in the annexed Table XLIII., the horse-powers required on (*a*) pure high-speed lathes, (*b*) ordinary carbon steel lathes, alongside of (*c*) the horse-powers we propose for the compromise lathes.

TABLE XLIII

Horse-powers for High-speed, Carbon, and Compromise Lathes.

Height of centres. .	6 in.	12 in.	18 in.	24 in.	30 in.	36 in.	48 in.	60 in.	72 in.
(*a*) High-speed steel	8·4	10·4	13·7	13·3	24	31·4	50	74	102
(*b*) Carbon steel . .	0·86	3·2	6·3	10	13	16	20	23	25
(*c*) Compromise . .	4·6	6·5	10	14½	20	28	46	70	100

TABLE XLIV

Table giving Depths of Cut, Widths of Traverse, and Area of Cut Proposed for Compromise Lathes from 6 in. to 6 ft. Height of Centres, also Proper Cutting Speed for those Cuts and the Net Horse-power Required.

Height of centres $= h$	6	9	12	15	18	21	24	27	30	36	42	48	54	60	66	72
Depth of standard cut $= \frac{h}{40}$	0·15	0·225	0·30	0·375	0·45	0·525	0·60	0·675	0·75	0·90	1·05	1·20	1·35	1·5	1·65	1·80
Width of standard traverse $= \frac{h}{160}$	0·0375	0·05625	0·075	0·0937	0·1125	0·1312	0·15	0·1687	0·1875	0·225	0·2625	0·30	0·34	0·375	0·4125	0·45
Area of standard cut $= \frac{h^2}{6400}$	0·00563	0·01265	0·0225	0·0352	0·0506	0·069	0·09	0·114	0·14	0·203	0·276	0·36	0·46	0·56	0·68	0·81
Cutting speed $= v = \frac{1}{2a} + 15$	104	54·5	37·2	29·2	24·9	22·25	20·56	19·39	18·57	17·465	16·81	16·39	16·086	15·893	15·736	15·55
Net horse-power $= 7{\cdot}8\ av$	4·56	5·38	6·52	8·0	9·82	11·97	14·4	17·22	20·3	27·65	36·15	46	57·7	69·4	83·5	98
Corresponding cutting speed on diameter $= 2h$ for 15 ft. per minute on diameter $= 2(h - \sqrt{h})$	25·3	22·5	21	20·2	19·6	19·17	18·87	18·55	18·35	18	17·74	17·5	17·35	17·2	17·1	17

TABLE XLV

Table of Values of Greatest and Least Spindle Revolutions, Compromise Lathe.

Height of centres $= h$	6	9	12	15	18	21	24	27	30	36	42	48	54	60	66	72
$N_g = \frac{2750}{h}$	458	306	229	183	153	131	114·5	101·8	91·6	76·4	65·5	57·3	51	45·8	41·7	38·2
$N_l = \frac{12 \times 15}{\pi \times 2(h - \sqrt{h})} = \frac{28{\cdot}7}{(h - \sqrt{h})}$	1·08	4·78	3·36	2·58	2·08	1·75	1·5	1·314	1·17	0·956	0·808	0·7	0·616	0·55	0·495	0·452
$\frac{N_g}{N_l} = \frac{2750(h - \sqrt{h})}{h \times 28{\cdot}7}$	56·8	64	68·2	71	73·6	75	76·4	77·5	78·3	80	81·1	82	82·8	83·4	84·2	84·6
$N_l = \frac{16 \times 12}{2\pi h} = \frac{30{\cdot}6}{h}$	5·1	3·4	2·55	2·04	1·7	1·456	1·275	1·133	1·02	0·85	0·729	0·638	0·567	0·51	0·464	0·425
$\frac{N_g}{N_l} = \frac{2750}{h} \times \frac{h}{30{\cdot}6} = 90$ throughout	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
$N_l = \frac{20 \times 12}{2\pi h} = \frac{38{\cdot}25}{h}$	6·38	4·25	3·19	2·55	2·125	1·82	1·595	1·415	1·275	1·062	0·91	0·798	0·71	0·638	0·58	0·532
$\frac{N_g}{N_l} = \frac{2750}{h} \times \frac{h}{38{\cdot}2} = 72$	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

The horse-power for the compromise lathe varies from being one-half of to equality with that assumed for the high-speed steel lathe.

With regard to *the lowest spindle speed required on the compromise lathe*, we are disposed to recommend that the slowest speed of spindle demanded be such as to enable finishing at 15 ft. per minute to be done upon work of diameter $(2h - \sqrt{h})$, *i.e.* work which can be swung over the saddle.

For carbon steel lathes a reference to Table XXXVI.—p. 69—will show that much lower speeds than this are supplied for the larger sizes of lathe, *e.g.* the speed varies from 20 ft. for a 6-in. lathe down to 4 ft. per minute for a 72-in. lathe. Such slow speeds can hardly be contemplated in lathes of the compromise variety.

If the above cutting or finishing speed be accepted for all sizes of lathe, we have the following formula for N_t

$$N_t = \frac{12 \times 15}{\pi \times 2(h - \sqrt{h})} = \frac{28{\cdot}7}{h - \sqrt{h}} \quad . \quad . \quad . \quad . \quad . \quad (66)$$

This quantity is tabulated in Table XLV., as also the values of N_g (and of $\frac{N_g}{N_t}$), as above determined. They have also been plotted on a base of height of centres in Fig. 34. On this figure are also shown values of N_g and N_t for a number of what may properly be called compromise lathes, as obtained from recent practice.

CHAPTER XVII

COMPROMISE LATHE—THE BELT—CONE ON SPINDLE

In the present chapter we propose to calculate the belt widths for a few typical cases of the compromise lathe, and to compare the results with those for high-speed steel and for carbon-steel lathes.

Compromise Lathe, Cone on Spindle.—We shall begin by working out the case of a lathe of, say, 12 in. height of centres, having the cone mounted on the spindle, to which it is coupled direct for the highest speeds of the latter.

The values of $$N_g = \frac{2750}{h} = \frac{2750}{12} = 229$$

of $$N_l = \frac{28{\cdot}7}{(h - \sqrt{h})} = 3{\cdot}36$$

and of $$\frac{N_g}{N_l} = 68{\cdot}2$$

have been already given in Table XLV., p. 89.

If we take $p = 2$, *i.e.* two speeds of the countershaft; $q = 4$, *i.e.* four cone steps; and $m = 3$, *i.e.* two back gears (or three different ways in which the cone may be coupled to the spindle); we have $s = pqm = 24$, or there are twenty-four different possible speeds of the spindle.

Hence, r is known from $$r^{s-1} = \frac{N_g}{N_l}$$

Thus we have $$\log r = \frac{\log \left(\frac{N_g}{N_l}\right)}{(s-1)}$$

and in this case $$\log r = \log \frac{68{\cdot}2}{23} = 0{\cdot}0797$$

or $$r = 1{\cdot}201$$

As for Case I., on p. 50, *ante*, we have

$$V_l = \frac{\pi A n_l}{60 \times 12} = \frac{\pi A N_g}{12 \times 60 r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (40)$$

Now, taking $A = 2h$ and $N_g = \frac{2750}{h}$, we obtain

$$V_l = \frac{\pi \times 2h \times 2750}{12 \times 60 \times h \times r^{2q-1}} = \frac{24}{r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (42\text{A})$$

It is therefore necessary to tabulate values of r^{2q-1} in order to compute

V_t, that speed of the belt when the pull upon it is greatest, and for which it must be designed.

Now $$\log r^{2q-1} = (2q - 1)\log r$$
$$= (2 \times 4 - 1) \times 0\cdot0797$$
$$= 0\cdot5579$$

Therefore $$r^{2q-1} = 3\cdot613$$

and $$V_t = \frac{24}{3\cdot613} = 6\cdot64 \text{ ft. per second}$$

Now the allowable tension, according to Gehrckens, as given for a double belt by equation 37, p. 48, is

$$P = \sqrt{D}\left(7\cdot5 + \frac{V}{10}\right)$$
$$= \sqrt{A}\left(7\cdot5 + \frac{6\cdot64}{10}\right)$$
$$= \sqrt{2h} \times 8\cdot2 = \sqrt{24} \times 8\cdot2$$
$$= 39\cdot95 \text{ lb. per inch of width}$$

If we allow double of this, or 80 lb., for the allowable tension per inch of width as being in better agreement with English practice, we finally obtain the required width of belt B from the expression

$$B = \frac{550\,\text{GHP}}{2PV_t}$$

The value of GHP for the "compromise lathe" may be taken from Table XLIII. or XLIV. (Chap. XVI.), and is 6·52 for a 12 in. lathe.

Thus $$B = \frac{550 \times 6\cdot52}{80 \times 6\cdot64} = 6\tfrac{3}{4} \text{ in.}$$

In Table XLVI, annexed, we have presented the results of similar calculations for lathes from 6 in. to 24 in. height of centres, and for the various numbers of double-back gears and cone steps specified in lines five and seven. The countershaft has two speeds throughout. The belt widths, given in line sixteen, are the least values that should be taken, being based upon an effective belt tension of double that recommended by Gehrckens for belts which drive continuously.

The values of B, the belt width, thus obtained, have been shown as continuous curves on a base of height of centres in Fig. 35. The marked effect of increasing the number of back gears in decreasing the belt width for any given size of lathe is again clearly shown. The dot and ring spots plotted in the same figure give the most recent results from practice.

This figure reveals at once how defective the belt drives ordinarily supplied are, for lathes of the compromise variety, when the cone is mounted on the spindle and drives it directly for the highest speeds. *Even with three double back gears* the sizes of belt ordinarily used on lathes from 6 in. to 10 in. centres would not be sufficient; and as three back gears are quite unusual on lathes below 18 in. centres, it is manifest that the sizes given for 12 in. to 18 in. centre lathes are also too small.

The belt widths plotted from practice seem to follow some such law as

$$B = \tfrac{2}{7}(h + 3)$$

but if this rule is used it must be remembered that the belt so designed cannot transmit the power required by our Table XLIV., *unless the cone is run at a much higher speed* than that to which it is limited by being coupled to the spindle directly for the highest speeds, *unless*, that is, *a pair of wheels is supplied between cone and spindle.*

It may be remarked that throughout the above calculations the diameter of the largest cone step has been taken—A = $2h$—equal to twice the height of centres. The validity of this rule is shown by the black dots plotted in Fig. 35, as they represent the values of the diameters of the largest cone steps from recent practice.

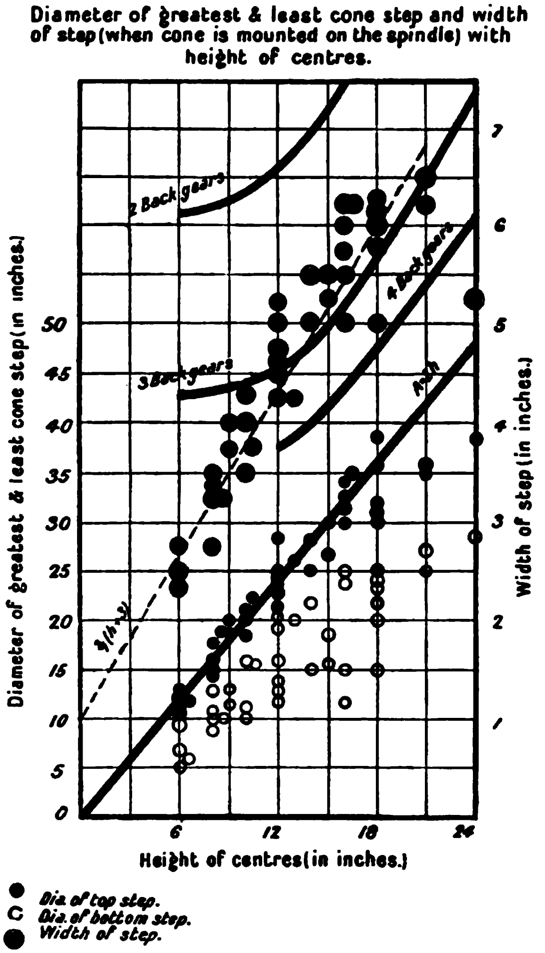


FIG. 35.

We may now institute a comparison between the belts required on a 12-in. centre lathe for high-speed steel, for carbon steel, and for both, as per compromise design just referred to.

Referring to Tables XXX. (*a* and *b*) and to Table XXXVIII., we may extract the data and results for the two former varieties for a 12-in. lathe, whilst in Table XLVI. we have the particulars for the compromise.

TABLE XLVI

Compromise Lathes. Belt Widths when the Cone is Mounted on the Spindle and directly coupled to it for the Higher Speeds. Two Countershaft Speeds. $A = 2h$. Belt Tension, Double of Gehrckens.

h	6			9			12			15			18			21		24		1
$N_g = \frac{2750}{h}$	458			306			229			183			153			131		114·5		2
$N_l = \frac{28·7}{h - \sqrt{h}}$	8·08			4·78			3·36			2·58			2·08			1·75		1·5		3
$\frac{N_g}{N_l} = 95·8\left(1 - \frac{1}{\sqrt{h}}\right)$	56·8			64			68·2			71			73·6			75		76·4		4
Number of back gears	2	2	3	2	2	3	2	3	4	2	3	4	2	3	4	3	4	3	4	5
m	3	3	4	3	3	4	3	4	5	3	4	5	3	4	5	4	5	4	5	6
q	3	4	3	3	4	3	4	3	3	4	3	3	4	4	4	4	4	4	4	7
s	18	24	24	18	24	24	24	24	30	24	24	30	24	32	40	32	40	32	40	8
$\text{Log } r = \dfrac{\log \frac{N_g}{N_l}}{s-1}$	0·103	0·0762	0·0762	0·1062	0·0786	0·0786	0·0797	0·0797	0·0632	0·0805	0·0805	0·0638	0·0812	0·0602	0·048	0·0605	0·0481	0·061	0·0485	9
r	1·268	1·192	1·192	1·277	1·199	1·199	1·201	1·201	1·157	1·203	1·203	1·158	1·206	1·149	1·117	1·149	1·117	1·151	1·118	10
r^{2q-1}	3·273	3·415	2·404	3·396	3·55	2·472	3·618	2·503	2·07	3·66	2·526	2·084	3·701	2·638	2·168	2·652	2·171	2·673	2·185	11
$V_l = \frac{24}{r^{2q-1}}$	7·34	7·02	10	7·07	6·76	9·7	6·64	9·6	11·6	6·56	9·5	11·5	6·5	9·1	11·05	9·06	11·05	9	11·0	12
$P = \sqrt{D}(7·5 + 0·1V_l)$ $= \sqrt{2h}(7·5 + 0·1V_l)$	28·54	28·43	29·46	34·8	34·67	35·9	39·95	41·4	42·4	44·59	46·2	47·3	48·9	50·46	51·6	54·43	55·76	58·2	59·6	13
2P	58			70			82			92			100			110		120		14
GHP	4·56			5·38			6·52			8·0			9·82			11·97		14·4		15
$B = \frac{550 \text{ GHP}}{2PV_l}$	5·9	6·16	4·3	6	6·25	4·35	6·6	4·5	3·75	7·25	5	4·15	8·3	5·94	4·7	6·6	5·4	7·35	6	16
$A = 2h$	12			18			24			30			36			42		48		17

Collecting them together they stand thus—

Comparison of Belt Drives for different Types of 12-in. Lathe. Cone on Spindle. Two Countershaft Speeds.

Type	High-speed steel.		Carbon steel.		Compromise type.	
Table number	XXX.		XXXVIII.		XLVI.	
Horse-power	10·4		3·16		6·52	
N_g and N_l	300 and 9·9		152 and 2·87		229 and 3·36	
$\frac{N_g}{N_l}$	30·3		53·3		68·2	
Number of cone steps	4		4		4	
Number of back gears	2	3	2	3	2	3
Number of speeds	24	32	24	32	24	32
r	1·160	1·116	1·189	1·137	1·201	1·147
r^{2q-1}	2·823	2·158	3·350	2·458	3·613	2·596
V_l	10·6	13·9	4·1	5·7	6·64	9·25
2P	84	88	73	73	82	82
B (double)	6½ in.	4¾ in.	5⅞ in.	4¼ in.	6¾ in.	4⅞ in.
A	24 in.	24 in.	21 in.	21 in.	24 in.	24 in.

The comparative results shown by this table may at first sight appear surprising. We find, for example—

(1) The lathe for carbon steel requires a width of belt almost as large for its 3·16 GHP as does the high-speed steel lathe for its 10·4 GHP! This is accounted for by the higher belt speed obtainable—when pulling the heaviest cut—in the latter than in the former case. The ratio of greatest to least spindle speed is much smaller for the high-speed than for the slow-speed lathe, and the number of spindle speeds being the same, r and r^{2q-1} is much smaller and V_l is much greater for the former than for the latter. The difference is only in very small measure due to the different belt tensions allowed.

(2) The compromise lathe requires a heavier belt than either of the others, because the range of possible spindle speeds and their ratio is greater; thus, with the same number of speeds, V_l is not much greater than for the carbon steel lathes, whilst the horse-power to be transmitted is more than double what they require.

As against the high-speed steel lathe the belt speed is reduced for the compromise lathe in a greater ratio than the horse-power is, consequently a wider belt is required.

One thing is clear from these results; and that is, that in lathes of this size, with cones mounted on the spindle, the belts, as now ordinarily supplied, with from one-half to two-thirds of these calculated widths are quite incapable of transmitting the power really required, unless they are strained so much beyond their safe working load as to be subject to frequent repairs and renewals. The purchaser's bill for stoppages in such lathes will be a large one, and he would do better to buy a lathe with a belt of the proper width to begin with.

CHAPTER XVIII

COMPROMISE LATHE—THE BELT—CONE OFF SPINDLE

Compromise Lathe, Cone off Spindle.—We may now take up the case of a larger size of lathe—say, one of 18 in. height of centres, of the "compromise" type, in which the cone is mounted on a back shaft, and a pair of wheels is interposed between it and the spindle.

The values of N_g, N_l, and $\frac{N_g}{N_l}$ are those given in Table XLV. If we again take $p = 2$, $q = 4$, and $m = 3$, we have $s = pqm = 24$.

Since—from Table XLV.—$\frac{N_g}{N_l}$ is 73·6, for $h = 18$ in., we have

$$\log r = \log\frac{73{\cdot}6}{23} = 0{\cdot}08116$$
$$r = 1{\cdot}206$$

Again we have, as in equation (45), p. 54, *ante*

$$V_l = \frac{\pi A R_o N_g}{12 \times 60 \times r^{2q-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

in which r^{2q-1} is known, being here equal to 3·701; N_g is known, being equal to 153; and we have only to fix upon the values of A and R_o. We shall take $A = 6\sqrt{h}$ for the diameter of the largest cone step, as this is the greatest diameter which is constructionally convenient. The spur and pinion (giving the "single-gear" ratio R_o) get to be too large if A is taken equal to $2h$, as in the last example. We shall also take $R_o = \sqrt{h}$, as was done before for carbon steel lathes (Case II., p. 79).

Fig. 36 gives evidence that our rule for A ($= 6\sqrt{h}$) is supported by recent practice (see curve drawn through full black dots).

We then have

$$V_l = \frac{\pi \times 6\sqrt{h} \times \sqrt{h} \times 2750}{720 \times r^{2q-1} \times h} = \frac{72}{r^{2q-1}} \quad . \quad . \quad . \quad . \quad (67)$$
$$= \frac{72}{3{\cdot}701} = 19{\cdot}45, \text{ for this case}$$

The allowable belt tension $P = \sqrt{A}\left(7{\cdot}5 + \frac{V}{10}\right)$ is here $P = \sqrt{25{\cdot}4}$ $(7{\cdot}5 + 1{\cdot}945) = 5{\cdot}04 \times 9{\cdot}5 = 47\tfrac{1}{2}$; and $2P = 95$, say.

Hence $B = \frac{550\,\text{GHP}}{2PV_l} = 550 \times \frac{9{\cdot}82}{95} \times 19{\cdot}45 = 2{\cdot}92$, or, say, 3 in.

In Table XLVII. the results of similar calculations for lathes from 12 in. to 72 in. height of centres are given for various values of q and $(m - 1)$. The countershaft has two speeds throughout.

The belt widths so obtained (see line 17) are the least values which should be taken, as the belt tension allowed is very high, considering the low belt speeds available.

These values are plotted as continuous curves, one for each value of the number of back gears, in Fig. 36, whilst the dot and ring spots give data collected from practice.

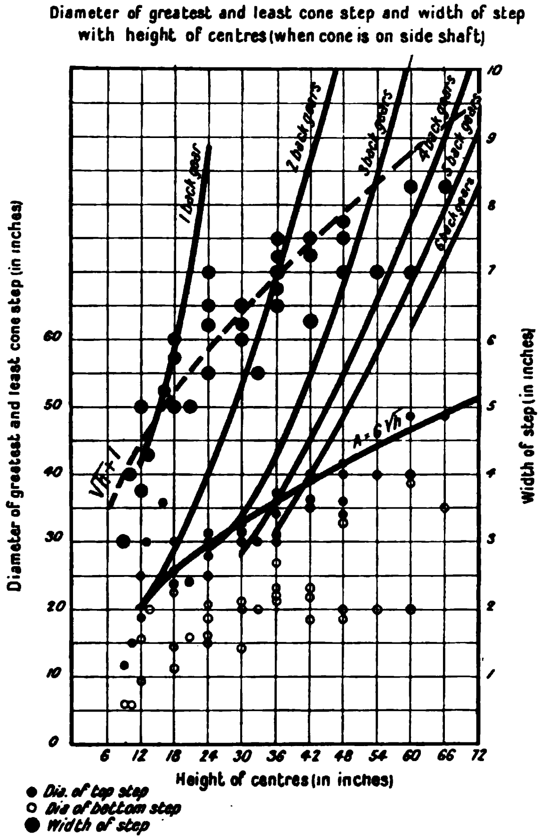


FIG. 36.

It will be found that the expression

$$B = 1 + \sqrt{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

gives widths of belt which agree with the average values of practice (*vide* dotted curve in figure). This formula will only give a belt sufficiently strong to transmit the horse-power we have postulated, provided that the number of back gears be fixed according to the law

$$\text{number of back gears} = (m - 1) = \frac{h}{18} \quad . \quad . \quad . \quad (69)$$

(The rule for carbon steel lathes usually adopted in practice is $(m - 1) = h/12$. The number of back gears is $h/12$.)

TABLE XLVII

Compromise Lathes. Belt Widths when Cone is Mounted on a Back Shaft, and a Pair of Wheels of Velocity Ratio $R_o = \sqrt{h}$ is interposed between it and the Spindle. Two Countershaft Speeds. $A = 6\sqrt{h}$. Belt Tension Double of Gehrckens.

h	12			15			18			21			24			1
$N_g = \frac{2750}{h}$	229			183			153			131			114·5			2
$N_l = \frac{28·7}{h - \sqrt{h}}$	3·36			2·58			2·08			1·75			1·5			3
$\frac{N_g}{N_l} = 95·8\left(1 - \frac{1}{\sqrt{h}}\right)$	68·2			71			73·6			75			76·4			4
Number of back gears	1	2	2	1	2	2	1	2	2	2	2	3	2	3	3	5
m	2	3	3	2	3	3	2	3	3	3	3	4	3	4	4	6
q	4	3	4	4	3	4	4	3	4	3	4	3	4	3	4	7
s	16	18	24	16	18	24	16	18	24	18	24	24	24	24	32	8
$\text{Log } r = \frac{\log \frac{N_g}{N_l}}{s - 1}$	0·122	0·1077	0·0798	0·1235	0·1088	0·0805	0·1245	0·11	0·0812	0·11	0·0815	0·0815	0·0822	0·0822	0·061	9
r	1·324	1·281	1·201	1·329	1·284	1·203	1·332	1·288	1·206	1·288	1·206	1·206	1·209	1·209	1·151	10
r^{2q-1}	7·145	3·455	3·619	7·319	3·499	3·66	7·439	3·548	3·701	3·548	3·719	2·556	3·761	2·576	2·673	11
$V_l = \frac{72}{r^{2q-1}}$	10·06	20·8	19·9	9·84	20·6	19·65	9·68	20·3	19·45	20·3	19·35	28·2	19·15	28	26·9	12
$P = \sqrt{6\sqrt{h}}(7·5 + 0·1V_l)$	38·8	43·5	43·1	40·8	46·1	45·5	42·7	48·1	47·6	49·8	49·4	54	51	55·7	55·1	13
2 P	85			90			95			100			105			14
GHP	6·52			8			9·82			11·97			14·4			15
$B = \frac{550\,\text{GHP}}{2PV}$	4·2	2·025	2·1	4·98	2·88	2·49	5·9	2·8	2·93	3·24	3·4	2·33	3·94	2·69	2·8	16
= nearest ¼in.	4¼	2	2	5	2½	2½	6	2¾	3	3¼	3½	2½	4	2¾	2¾	17
$R_o = \sqrt{h}$	3·46			38·7			4·24			4·58			4·9			18
$A = 6\sqrt{h}$	20·78			23·24			25·4			27·5			29·4			19

TABLE XLVII—*continued.*

Compromise Lathes. Belt Widths when Cone is Mounted on a Back Shaft, and a Pair of Wheels of Velocity Ratio $R_o = \sqrt{h}$ is interposed between it and the Spindle. Two Countershaft Speeds. $A = 6\sqrt{h}$. Belt Tension double of Gehrckens.

h	30			36			48			60			72			1
$N_s = \frac{2750}{h}$	91·6			76·4			57·3			45·8			38·2			2
$N_t = \frac{28·7}{h - \sqrt{h}}$	1·17			0·956			0·7			0·55			0·452			3
$\frac{N_s}{N_t} = 95·8\left(1 - \frac{1}{\sqrt{h}}\right)$	78·3			80			82			83·4			84·6			4
Number of back gears	3	4	4	3	4	5	3	4	5	4	5	6	4	5	6	5
m	4	5	5	4	5	6	4	5	6	5	6	7	5	6	7	6
q	4	3	4	4	4	4	4	4	4	4	4	4	4	4	4	7
s	32	30	40	32	40	48	32	40	48	40	48	56	40	48	56	8
$\text{Log } r = \frac{\log \frac{N_s}{N_t}}{s-1}$	0·0612	0·0654	0·0486	0·0614	0·0488	0·0405	0·0618	0·049	0·0407	0·0493	0·0409	0·035	0·0495	0·041	0·035	9
r	1·152	1·162	1·119	1·152	1·119	1·097	1·153	1·119	1·098	1·120	1·098	1·084	1·120	1·099	1·084	10
r^{2q-1}	2·681	2·123	2·189	2·69	2·196	1·921	2·708	2·203	1·927	2·214	1·938	1·758	2·221	1·936	1·758	11
$V_t = \frac{72}{r^{2q-1}}$	26·85	33·9	32·9	26·8	32·8	37·5	26·6	32·7	37·4	32·5	37·3	41	32·4	37·2	41	12
$P = \sqrt{6\sqrt{h}}(7·5 + 0·1V_t)$	58·3	62·4	61·85	61·1	64·7	67·5	65·45	69·4	72·4	73·4	76·6	79·2	76·6	80·1	82·7	13
2 P	120			130			140			150			160			14
GHP	20·3			27·65			46			69·4			98			15
$B = \frac{550 \text{ GHP}}{2PV}$	3·46	2·74	2·83	4·36	3·56	3·12	6·8	5·53	4·83	7·83	6·82	6·22	10·4	9·05	8·32	16
= nearest ¼ in.	3½	2¾	3	4½	3½	3	6¾	5½	4¾	7¾	6¾	6¼	10½	9	8½	17
$R_o = \sqrt{h}$	5·48			6			6·93			7·75			8·5			18
$A = 6\sqrt{h}$	32·8			36			41·6			46·5			51			19

The following Table (XLVIII.) gives the belt widths from Table XLVII. in a collected form, and shows their dependence upon both size of lathe and number of back gears in a convenient form.

TABLE XLVIII

Belt Widths selected from Table XLVII. for Compromise Lathes with Cone off Spindle, Two Countershaft Speeds, and Three or Four Cone Steps.

$h =$	12	18	24	36	48	60	72	$m-1$
One back gear . . .	4¼	6	8	—	—	—	—	1
Two ,, . . .	2	3	4	7	10½	—	—	2
Three ,, . . .	—	—	2¾	4½	6¾	10¼	—	3
Four ,, . . .	—	—	—	3½	5½	7¾	10½	4
Five ,, . . .	—	—	—	3	4¾	6¾	9	5
Six ,, . . .	—	—	—	—	—	6¼	8½	6

A simple and roughly correct formula for the belt widths of these compromise lathes, with cone off spindle, is found to be

$$B = \frac{i}{2g}\text{GHP} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

where i = number of tools taking standard cut;

$g = (m-1)$ = number of compound gears supplied;

$\text{GHP} = \left(3{\cdot}9 + \frac{h^2}{55}\right)$ = gross horse-power required.

A still better approximation is obtained by writing

$$B = \frac{i}{2g}\left(5 + \frac{h^2}{60}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

We may usefully inquire what the highest countershaft speeds will be, and what speeds through the air the main shaft-countershaft belt will have, when we use the rules $R_o = \sqrt{h}$, $A = 6\sqrt{h}$, and $N_g = 2750/h$, as we have just been doing.

We have obviously (see Fig. 37)

$$c_1 = \frac{E}{A}R_oN_g$$

or

$$c_1 r^{\sqrt{q-1}} = R_oN_g$$

i.e.

$$c_1 = \frac{R_oN_g}{r^{\sqrt{q-1}}} = \frac{\sqrt{h} \times 2750}{r^{\sqrt{q-1}} \times h} = \frac{2750}{\sqrt{h}\,r^{\sqrt{q-1}}}$$

Now, throughout Table XLVII., for $s = 24$ we may take r constant $= 1{\cdot}2$, and with $q = 4$, $r^{\sqrt{q-1}} = 1{\cdot}317$.

Therefore

$$c_1 = \frac{2088}{\sqrt{h}}$$

But if X is the diameter of the smaller countershaft pulley, and U is the main-belt speed in feet per minute, we have, taking $X = 5\sqrt{h}$

$$U = \frac{5\sqrt{h}\pi c_1}{12} = \frac{5\sqrt{h}\pi}{12} \times \frac{2088}{\sqrt{h}} = \frac{5 \times 2088\pi}{12} = 2730 \text{ ft. per minute, } \textit{constant for all sizes of lathe}$$

With $Z = 48$ in., we shall have $u = \frac{2730}{4\pi} = 218$; *i.e.* the main-shaft speed being 218 revolutions per minute, 4-ft. diameter main driving pulleys will be required to give the belt this constant speed for any size of lathe.

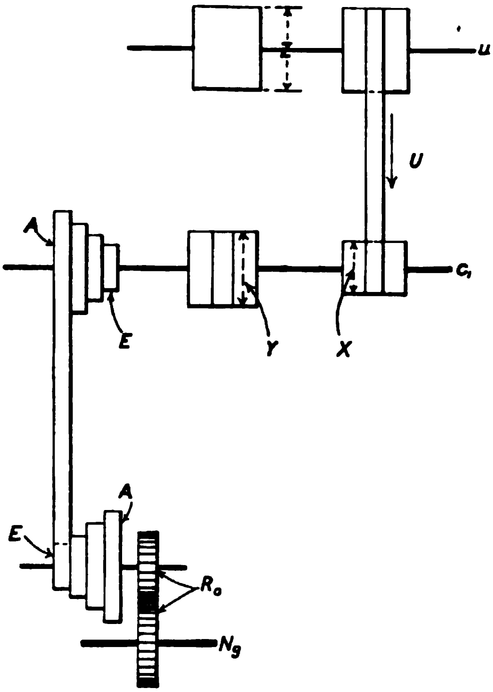


FIG. 37.

Compromise Lathe, Cone off Spindle, One Countershaft Speed.—If, for the larger sizes of lathe, *only one countershaft speed* is to be supplied, we proceed as follows:—

N_g and N_l are as before. Taking *e.g.* $h = 36''$, $p = 1$, $q = 5$, $(m - 1) = 4$, we have $s = pqm = 1 \times 5 \times 5 = 25$ speeds. $\frac{N_g}{N_l} = 80$. Therefore $r = 1{\cdot}2$.

Now, if c is the countershaft speed, we have $n_l = c\frac{E}{A}$ and $n_g = R_oN_g$

$= c\frac{A}{E}$

therefore $n_l = \frac{R_oN_g}{r^{q-1}}$

for $\frac{A}{E} = r^{\frac{q-1}{2}}$

But $$V_l = \frac{\pi A n_l}{60 \times 12} = \frac{\pi A R_o N_g}{720x^{q-1}} = \frac{\pi \times 6\sqrt{h}\sqrt{h} \times 2750}{720r^{q-1}h} = \frac{72}{r^{q-1}} \quad . \quad (72)$$

In the present case $r^{q-1} = 2{\cdot}078$; therefore $V_t = 34{\cdot}7$. The allowable tension is 66·6 and $2P = 130$.

Hence $$B = \frac{550\,\text{GHP}}{2PV_t} = \frac{550 \times 27{\cdot}65}{130 \times 66{\cdot}6}$$

or $$B = 3\tfrac{1}{2}\text{ in.}$$

Table XLIX. gives similarly obtained values for lathes of 3 ft., 4 ft., 5 ft., and 6 ft. centres.

TABLE XLIX

Compromise Lathes. Belt Widths as in Table XLVII., but with One Countershaft Speed only.

h	36		48		60		72	
$N_g = \frac{2750}{h}$	76·4		57·3		45·8		38·2	
$N_t = \frac{28{\cdot}7}{(h - \sqrt{h})}$	0·956		0·7		0·55		0·452	
$\frac{N_g}{N_t} = 95{\cdot}8\left(1 - \frac{1}{\sqrt{h}}\right)$	80		82		83·4		84·6	
Number of back gears	4	5	5	6	6	7	7	8
m = number of ways of coupling to spindle	5	6	6	7	7	8	8	9
q = number of cone steps	5	5	5	5	5	5	5	5
s = number of speeds	25	30	30	35	35	40	40	45
$\text{Log } r = \frac{\log \frac{N_g}{N_t}}{s-1}$	0·0794	0·0656	0·066	0·0563	0·566	0·0493	0·0495	0·0438
r	1·2	1·163	1·164	1·139	1·140	1·12	1·12	1·106
r^{q-1}	2·078	1·83	1·837	1·68	1·685	1·575	1·578	1·497
$V_t = \frac{72}{r^{q-1}}$	34·7	39·4	39·2	42·9	42·7	45·7	45·6	48·1
$P = \sqrt{6\sqrt{h}(7{\cdot}5 + 0{\cdot}1V_t)}$	66·6	68·6	73·7	76·1	80·3	82·4	86	87·8
$2P$	130		150		160		170	
GHP	27·65		46		69·4		98	
$B = \frac{550\,\text{GHP}}{2PV_t}$	3·37	2·97	4·3	3·93	5·6	5·22	6·95	6·6
= to nearest ¼	3½	3	4¼	4	5½	5¼	7	6½
$R_o = \sqrt{h}$	6		6·93		7·75		8·5	
$A = 6\sqrt{h}$	36		41·6		46·5		51	

In conclusion we may institute a comparison between high-speed, low-speed, and compromise lathes, *cone off spindle*, similar to what was made for cone on spindle.

The results of the comparison lead to remarks of a very similar character to those made about cases in which the cone was mounted on the spindle. These may be said to hold almost in their entirety, *mutatis mutandis*, with regard also to lathes with cones mounted on a back shaft, as represented by the annexed table:—

Comparison of Belt Drives for Three Types of 18-*in. Lathe. Cone off Spindle. Two Countershaft Speeds.*

Type	For high-speed steel.		For carbon steel.		Compromise.	
From Table	XXXII.		XLI.		XLVI.	
Horse-power	13·7		6·3		10	
N_g and N_t	200 and 3·9		102 and 1·7		153 and 2·1	
$\frac{N_g}{N_t}$	51·3		60		73	
Number of steps	4		4		4	
Number of gears	2	3	2	3	2	3
Number of speeds	24	32	24	32	24	32
r	1·187	1·133	1·195	1·141	1·206	1·149
r^{2q-1}	3·323	2·396	3·476	2·519	3·701	2·638
V_t	24	30	14	19	20	28
2P	120	120	90	95	96	106
B	3 in.	2½ in.	2⅞ in.	2 in.	3 in.	2 in.
A	31½ in.		26 in.		26 in.	
R_o	3·75	3·54	4·24		4·24	

The small belt width by means of which the required power can be obtained for this size of lathe, when the cone is mounted upon a back shaft, is clearly brought out by the comparison of this table with the corresponding one presented in Chapter XVII.

CHAPTER XIX

THE FAST HEADSTOCK, GEAR ARRANGEMENTS, CONE ON SPINDLE

We propose now to describe the principal designs of geared headstock driven by a cone mounted on the spindle which are in recent common use. We begin with Type I., the most usual arrangement for small lathes—that shown in Fig. 38. Here the cone and the pinion A are mounted together on a sleeve which runs freely on the spindle, but which is also capable of being locked to it. The wheels B and C are keyed upon a sleeve excentrically mounted upon a back shaft, and are put into and out of gear by rotating the shaft. Sometimes the wheels are keyed upon the back shaft itself, and are shifted along it to be put in and out of gear. This method requires a longer headstock than the other, and is therefore not so common. The front driving wheel D is keyed to the spindle. It also carries a lock bolt, and a recess in a flange on the end of the cone is made to receive the same. When the wheels B and C are out of gear, and the wheel D is locked to the cone, the spindle is driven directly by it, and has as many direct-driven speeds as there are cone steps. When the lock bolt is freed from the cone and the drive is through the back gear, the gear ratio is obviously $\frac{A}{B} \times \frac{C}{D}$, and there are as many back-gear speeds as there are cone steps for each countershaft speed.

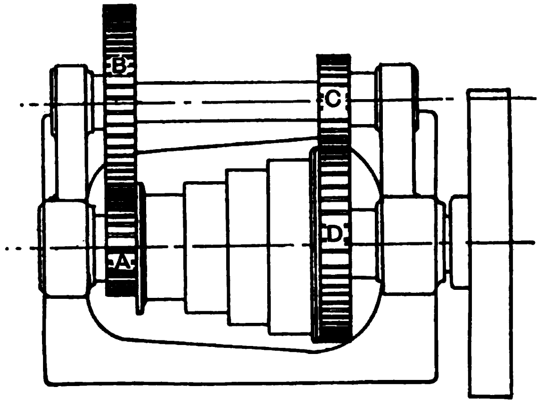


Fig. 38.

A clutch frequently takes the place of the lock bolt. It is usually a claw coupling keyed to the spindle and sliding along it so as to gear into either the cone or the wheel D, according as it is desired to run direct or through the back gear. This arrangement, though more expensive than the lock bolt, is much more convenient, and is also safer. In the case of lathes with headstocks of Type I., the total number of spindle speeds is twice the number of cone steps for each countershaft speed, or $mpq = 2pq$; where p and q are the numbers of countershaft speeds and cone steps respectively, and m is the number of different ways the cone can be coupled to the spindle.

TYPE II.—In Fig. 39 an arrangement is shown quite similar to that of Fig. 38, but with the provision of two back gears instead of one. This is simply effected by securing an additional pinion E on the cone sleeve, and a wheel—of equal size in the illustration—F on the back sleeve. We then have the two double-gear ratios—

$$\frac{E}{F} \times \frac{C}{D}, \text{ and } \frac{A}{B} \times \frac{C}{D}$$

The number of spindle speeds is three times the number of cone steps for every countermotion speed. For $pqm = pq \times 3$, as there are three different ways in which the cone can be coupled to the spindle.

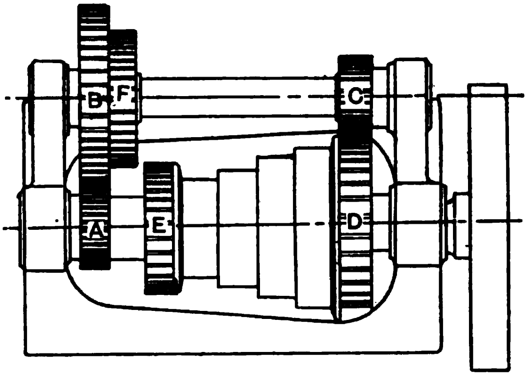


FIG. 39.

TYPE III.—In order to reduce the torque on the spindle, the front driving wheel is frequently combined with the face-plate. Such an arrangement is shown in Fig. 40, where the gears are otherwise exactly as described for Type II. The wheels B, F and C slide upon feathers on the back shaft for the purpose of engagement or disengagement. The number of spindle speeds is here again $3pq$. In some examples of this type the pinion and wheel A and B are brought to the front of the cone. (See *The Engineer*, "Machine Tool Supplement," November 4, 1904, high-speed lathe, by Jos. Buckton and Co.)

An additional back gear can obviously be obtained for every pair of wheels added to the tail end of the headstock similar to A, B and E, F of Fig. 40. Such a plan is, however, seldom resorted to, as the headstock becomes unduly long and the wheels are not employed to the best advantage, a larger number of gears being obtainable with the same number of pairs of wheels if arranged differently. Such a preferable arrangement is shown in—

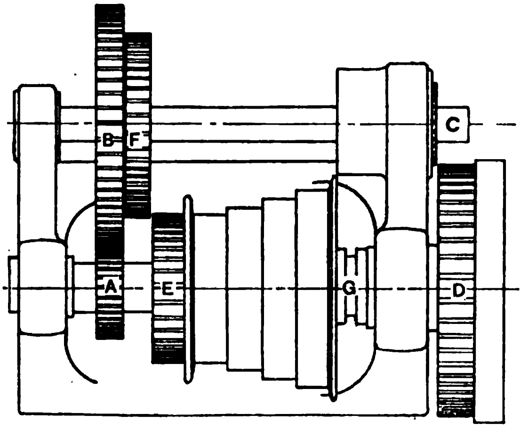


FIG. 40.

TYPE IV., and is pictured in Fig. 41. To obtain three double gears in the same way as two were provided in Types II. and III. four pairs of wheels would be necessary. If, however, the four pairs of wheels be fitted as shown in Fig. 41, four double back gears may be secured instead of three. Here A and E are fixed to the cone sleeve, whilst H is keyed to the spindle. Thus the drive may be through A, B, G, H, or E, F, G, H. But by withdrawing

G to the right and putting C in gear with the face-plate wheel D we obtain two new drives, viz. A, B, C, D, and E, F, C, D.

When two pinions such as C and G are to slide together, so that one goes into whilst the other goes out of gear, if the numbers of teeth in the two wheels with which they gear be not commensurable, the width of two wheels plus clearance must be allowed. Otherwise much time may be lost in bringing the space of the wheel opposite the tooth of its pinion. With an excentric sleeve arrangement on the back shaft this point need not be considered.

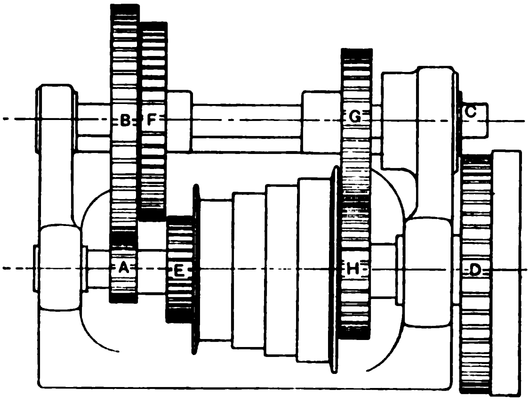


Fig. 41.

All the gears should be interlocking to prevent accidents. The added convenience in handling and the removal of the possibility of the lathe standing for several days whilst a broken gear is being replaced will much more than repay the additional thought required in the designing-room and the extra expense in manufacture.

The running of wheels idle should be avoided. Some makers apparently think this a matter of small importance; but there is needless wear, expenditure of power, and noise. As has been already insisted upon, the cone, even when mounted on the spindle, should never be coupled directly to it, but only through the intermediary of one or more pair of wheels. An arrangement of this kind by Messrs. Herbert, of Coventry, was illustrated in Fig. 2 of *The Engineer* Supplement, November 4, 1904.

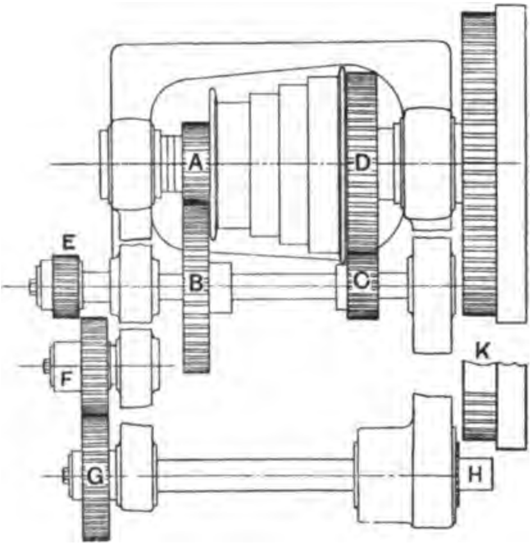


Fig. 42.

Type V.—Fig. 42 shows a common way of obtaining a double and treble back gear. By a treble gear is meant one in which the velocity ratio is increased in three steps, or by the interposition of three pairs of wheels between cone and spindle. Thus, in the present example, C being drawn out of gear with D, and E slid into gear with F, we have the gear ratio $\frac{A}{B} \times \frac{E}{G} \times \frac{H}{K}$; the idle wheel F being merely inserted to give the spindle the proper direction of rotation. To run in double gear, C is put into and E out of gear with their respective wheels, and the drive is through A, B, C, D; D being keyed to the spindle, and pinion H withdrawn.

This arrangement has the bad feature, in common with previously described types, that when running direct the cone is locked to D; so that the belt speed is limited to that due to the relatively small number of revolutions per minute permissible for the lathe spindle, and the belt becomes unmanageably wide for high-speed steel. The necessity of inserting the idle wheel F is also reprehensible. It has usually to be of small diameter to enable pinion H to mesh with wheel K, whilst at the same time obtaining the proper ratio between wheels E and G; and being both a driving and a driven wheel, it is liable to excessive wear and irregular motion. Being also usually overhung, it wears bell-mouthed, and very soon wobbles most objectionably. Another way of obtaining the necessary change of direction of rotation for a treble back gear is to have a crossed belt to the countershaft. This is unusual in this class of lathe, which is generally provided with two countermotion speeds. The addition of a third set of pulleys would make a somewhat cumbrous arrangement.

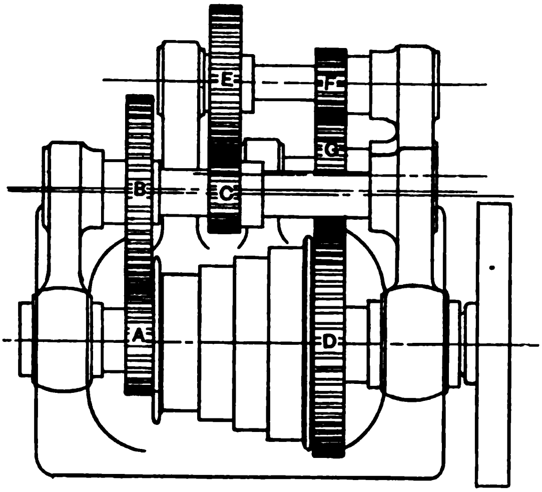


Fig. 43.

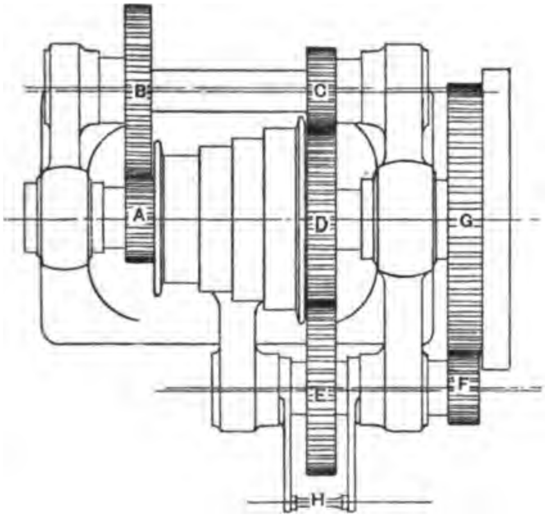


Fig. 44.

Some continental makers place the intermediate wheel between H and K, and give the shaft carrying it a bearing in both cheeks of the headstock. The irregularity of motion is, however, then liable to be even more pronounced, as the pitch is coarser the slower the speed of revolution.

Type VI.—Fig. 43 shows such an arrangement as carried out by a German firm.

It gives the same number of speeds—viz. $3pq$—as Type No. V., but with wheels fewer in number by two. The cone is locked to the spindle when running direct. When running double gear A drives B, and C is made to gear with D, G being withdrawn. When running treble gear—as shown—A drives B, C drives E, and F, through the idle wheel G, drives D.

TYPE VII.—Fig. 44 shows a recent German patented arrangement of double and treble geared headstock.

An independent plate keyed to the spindle is placed between the cone and the wheel D, both of which run loose on the spindle. The arrangement is such that, (1) either the cone, or (2) D, can be locked to the plate, or (3) the cone and wheel D can be locked together, leaving the plate and spindle to run at a different speed, or (4) D can run free of both cone and lock-plate. To run direct the cone is locked to the plate, and E and F are then disengaged by means of two excentric bushes coupled together by the handle H. For driving through the first double gear A, B, C, and D, the plate and wheel D are locked together, the cone being free. To run through a second double gear, the cone and D are connected together—without the plate being fastened to them—and the drive is D, E, F, and G; the wheels B and C being thrown out of gear by an excentric shaft arrangement. When running treble gear D is freed from both cone and lock-plate, and A drives B; C, through the intermediate D, drives E, and F drives G. Number of spindle speeds is $4pq$.

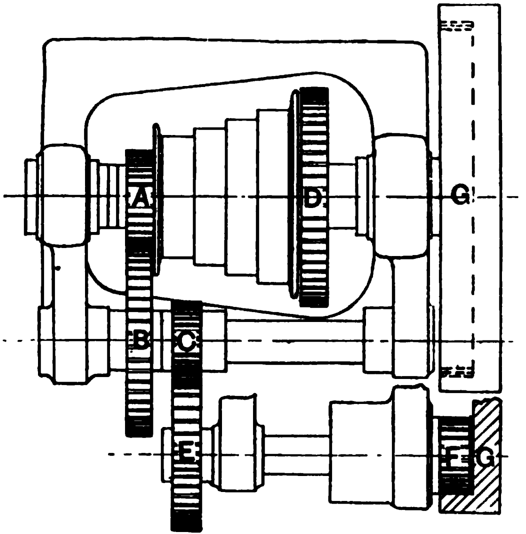


FIG. 45.

By superposing Types II. and VII., *i.e.* by adding another pair of wheels at the tail end of the cone, three double and two treble gears may be obtained with nine wheels; and the number of speeds will be $6pq$.

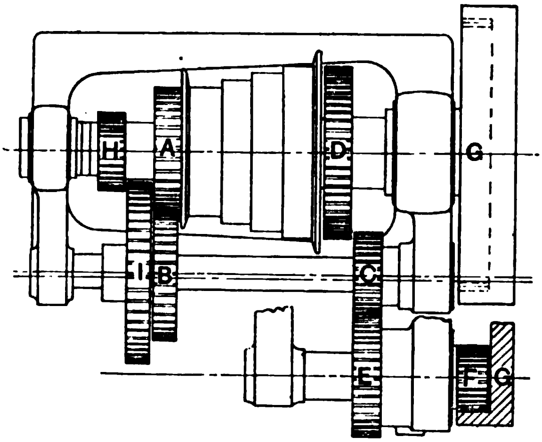


FIG. 46.

TYPE VIII.—To get rid of the troubles arising from intermediate wheels, it is usual to make the face-plate internally geared, and so to obtain the proper direction of rotation of the spindle. Fig. 45 shows such an arrangement.

When running single geared, B is disengaged by an excentric shaft, F is withdrawn, and the cone is locked to the spindle.

When running double geared, A drives B, and C is made to gear with D, F being drawn out.

When running treble geared, A drives B, C drives E, and F drives G; the cone being free.

Internal gears work, as is well known, more smoothly than externally geared wheels, but are somewhat more expensive to cut.

Type IX.—Fig. 46 shows a similar arrangement to Fig. 45, and bears the same relation to Type VIII. as Type II. does to Type I. By fitting another pair of wheels at the tail end of the cone, two double and two treble gears are secured. Thus we have: (1) A, B, C, D; (2) H, I, C, D; (3) A, B, C, E, F, G; and (4) H, I, C, E, F, G. Number of spindle speeds, *5pq*.

Type X.—Fig. 47 shows the arrangement of gearing in a recent German lathe, which is intended to use high-speed steels for roughing cuts, and to finish with ordinary carbon steels at slow speed. It allows of one double and one treble gear without the face-plate wheel; and one double and two treble gears with the face plate. A locking plate is provided between cone and wheel D, similar to what has been already described under Type VII.

D is free on the spindle, but can be locked to the cone or the driving plate. Wheel B and the clutch are keyed upon a sleeve running freely upon an excentric shaft; H and C are mounted freely on the same sleeve; whilst J, E, and F are keyed to the face-plate pinion shaft, being, however, free to slide along the same.

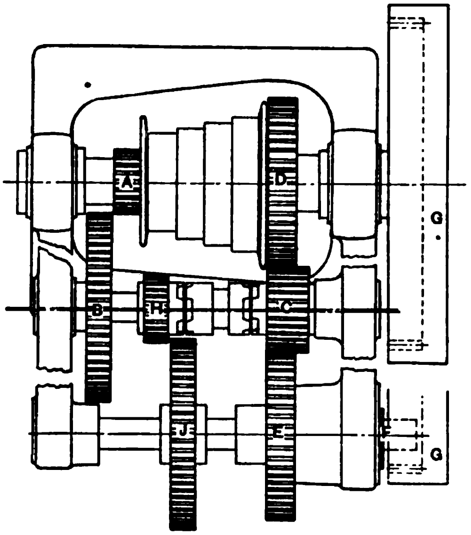


Fig. 47.

The drives obtainable are then as follows:—

(1) Double gear through D and spindle: A, B, C, D. Clutch engaged with C, and E withdrawn to the left. D locked to plate.

(2) Treble gear through spindle: A, B, H, J, E, C, D. Clutch geared with H; D locked to plate.

(3) Double gear through face-plate wheel: D, C, E, F, G. In this case the wheel D is locked to the cone, but is free from the plate.

(4) Treble gear through face plate: A, B, H, J, F, G. D being free of both cone and driving plate; clutch engaged with H, and E withdrawn.

(5) Second treble gear through face plate: A, B, C, E, F, G. D is freed from both cone and driving plate, and the clutch is in gear with C.

A careful examination of this arrangement seems to indicate that it is impossible to get all the speeds in continued geometrical progression. Whatever values are taken for the gear ratios, a gap in the curve of speeds seems to be inevitable. It is also bad practice to drive from wheel to pinion, as the speed of revolution is increased only to be again diminished.

Type XI. is only a slight modification of Type IX. If another pinion be added to the back shaft, and a mating wheel on the treble gear shaft in

Fig. 46, we obtain the arrangement of Fig. 48. In this case two double and four treble gears can be obtained without using D as a driver.

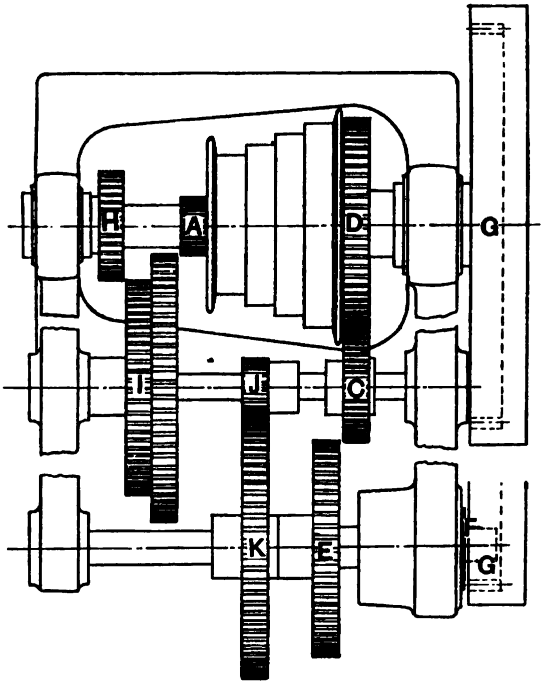


FIG. 48.

TYPE XII.—Fig. 49 shows an arrangement frequently used for double- and treble-geared lathes, the internally geared spur wheel being mounted on the treble-gear shaft. In this way two double and two treble gears are secured as thus:—

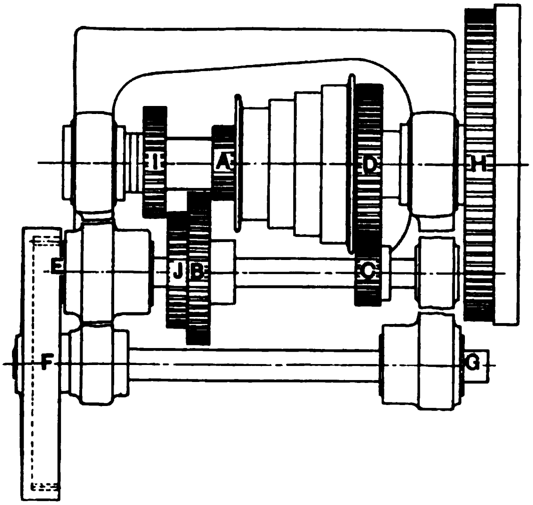


FIG. 49.

(1) First double gear, I, J, C, D.
(2) Second double gear, A, B, C, D.
(3) First treble gear, I, J, E, F, G, H.
(4) Second treble gear, A, B, E, F, G, H.

A disadvantage of this plan is that as the wheel F must of necessity be overhung, the treble gear shaft is of unusual length, and to be able to transmit the whole torque without undue twist it must be made very stiff.

CHAPTER XX

THE FAST HEADSTOCK—GEAR ARRANGEMENTS—CONE OFF SPINDLE

In our last chapter all the gear arrangements we illustrated had the objectionable feature that the cone was mounted upon the spindle, and was capable of being directly coupled to it when the highest spindle speeds are required.

In the present number we proceed to consider cases where at least one pair of wheels is interposed always between cone and spindle. The effect of this is that a much higher belt speed, and therefore a much lighter belt, can be used; or, for any given width of belt, the lathe becomes much more powerful.

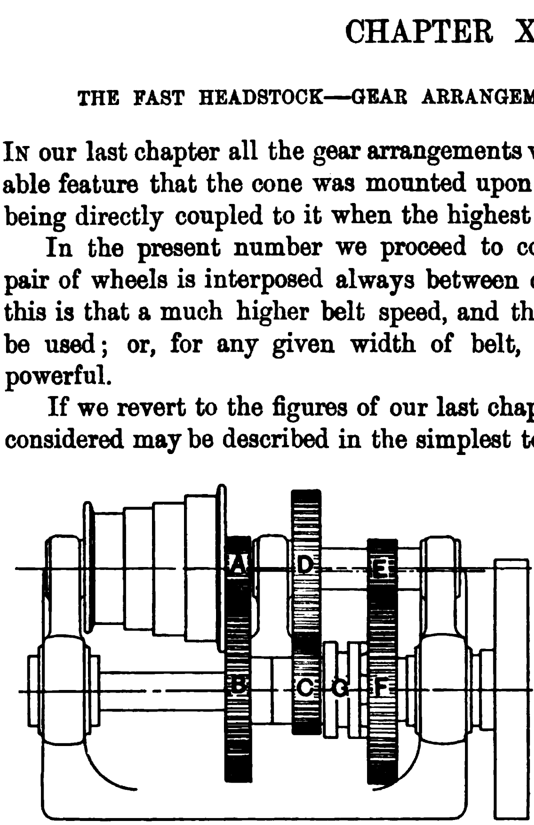


Fig. 50.

If we revert to the figures of our last chapter, it is clear that all the types considered may be described in the simplest terms by saying that they consist essentially of one or more double gears, one or more treble, combined with one or more double gears, and so on. In the present case, therefore, we ought to be able to present a cone-off spindle arrangement corresponding to every type of cone-on spindle gear shown there, by simply interposing a pair of wheels, of ratio R_0, anywhere between the cone and the spindle; and, further, that by having a reverted train of two, or any even number of, pairs of wheels the cone may even be mounted on the spindle, provided that for driving purposes it cannot be directly coupled to it. As an instance, we may refer to Fig. 41. In that type, if the wheel H be coupled to the cone and free on the spindle for the highest spindle speed, the cone revolutions will be greater than the spindle revolutions in the proportion of a ratio $R_0 = \frac{D}{C}$, assuming H and G are equal.

Considering first types I. to IV., which comprise headstocks with one or more double gears only, we may present the arrangements shown in Figs. 50 and 51, in which, by the interposition of gears $\frac{A}{B}$, ratio $\frac{B}{A} = R_0$, between cone and spindle, we obtain from Type I. one single and one treble gear.

In Fig. 50—a detailed arrangement similar to which was illustrated in

The Engineer Supplement of November 4, 1904, as made by the Miley Machine Tool Co.—the cone pulley and pinion A revolve in one piece on the back shaft, whilst the wheel B, pinion C, and clutch G are keyed to a sleeve rotating freely on the spindle. The clutch G is free to slide along the sleeve and engage with the wheel F, as when running in single gear. D and E are fixed to a sleeve which runs freely upon an excentric back shaft, so that they can be thrown out of gear with C and F when the clutch is in. For running treble geared the clutch is withdrawn, D and E are engaged with C and F, and the drive is through A, B, C, D, E, and F. $s = mpq = 2 \times pq$.

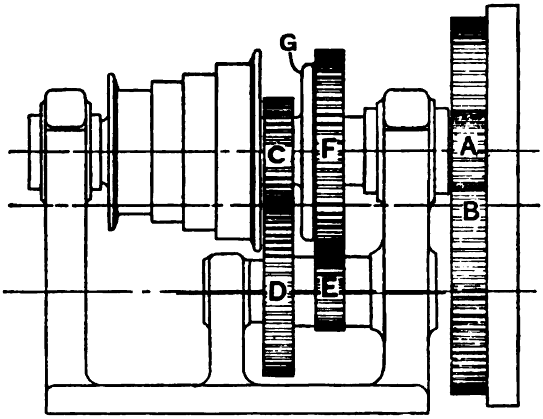


FIG. 51.

Fig. 51 shows a design that gives the same number of spindle speeds, with one single and one treble gear, the only difference being that the single-gear wheels are placed at the face-plate end instead of the cone end of the train of wheels. The cone, pinion C and locking plate G freely revolve as one piece upon the side shaft, whilst the wheel F and pinion A are keyed to it. D and E are keyed upon a sleeve running freely upon an excentric shaft. To run in single gear G and F are locked together, and D and E withdrawn, and the cone driving the lock plate drives wheel B through pinion A, which is always in gear. When working in treble gear the lock plate is freed, and the drive is through C, D, E, F, A, and B.

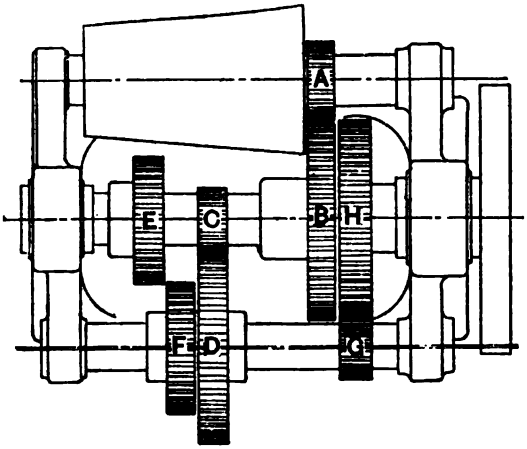


FIG. 52.

An obvious defect of the design of Fig. 51 is that, as the pitch of the teeth of A and B must be rather coarse to enable them to transmit the heavy load at the slower speeds, they are not so suitable for obtaining smooth running when working at the high speeds of the single gear. This defect is not found in the arrangement of Fig. 50, where the single gear is close to the cone.

Fig. 52 shows an arrangement where a tapered cone, having a curved surface, and mounted on a side shaft, gives motion to the spindle through the wheels A and B. On the spindle is a sleeve which is free to rotate independently thereof, and which carries the wheels B, C, and E all keyed to it.

The wheel H is keyed to the spindle itself, and is provided with a lock bolt, which fits the recesses in wheel B. F, D, and G are mounted on a sleeve, and can be disengaged or engaged by means of an excentric shaft. When running single gear the wheel B is locked to H, the back gear being disengaged; in treble gear the wheel B is freed from H, and drives the spindle through A, B, E, F, G, H, or A, B, C, D, G, H. With this arrangement it is possible to get any speed between the limits of the gears, and the difficulty of shifting a belt upon a stepped cone is avoided. The mating cone can be placed at the rear of the lathe and below the other.

An arrangement, by Messrs. J. Lang and Sons, by which a continuously variable speed can be obtained, was shown in the "Machine Tool Supplement" of *The Engineer* of November 4, 1904. Here two cones were caused to open or close, whilst a corresponding pair on a second shaft were caused to close or open. The belt drives, however, altogether by the edges, and for this reason the design is unsuitable for very high powers. It is only by having a high speed of strap and by gearing down to the spindle that a heavy cut can be taken with this arrangement.

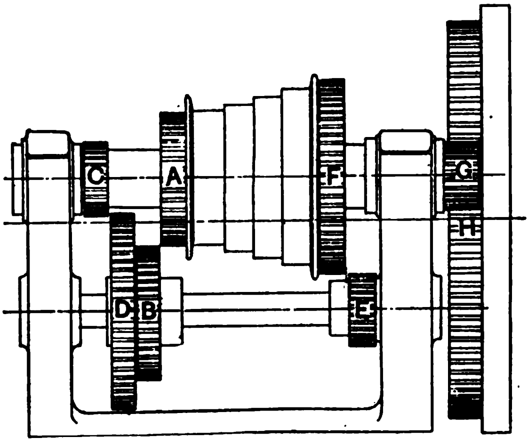


Fig. 53.

Sketch No. 53 shows an arrangement, used by several firms, capable of giving one single and two treble gears. The cone and pinions A and C run together as one piece freely upon the shaft, whilst wheel F and pinion G are keyed to it. The wheels D, B, and E are keyed upon the second-motion shaft, but free to slide along the same. When running single gear wheel F is locked to the cone, and the drive is through G and H; D, B, and E being disengaged. In treble gear the wheel F is freed from the cone, and we have: first treble gear, A, B, E, F, G, H; second treble gear, C, D, E, F, G, H.

It is a common continental practice to fix a second face-plate pinion on the second motion shaft. The countershaft is then provided with a reversing motion; and in this way two double gears are added to the above two treble gears. Suppose such a pinion to exist (I), we should have: first double gear, A, B, I, H; second double gear, C, D, I, H.

In Fig. 54 this idea (the design of Messrs. Sharp, Stewart, and Co.) is further extended, and one single gear, two double, and two treble gears are secured; whilst dispensing with lock-plates, clutches, and sleeves. At the same time a third-motion shaft is provided, and each shaft fitted with a face-plate pinion; but by this means the wear is well subdivided. The cone and wheels A, C, and G are keyed to the first-motion shaft, and rotate with it, thus preventing any possibility of wearing flats on the shaft, as is likely to happen when the cone, etc., run freely upon it. D, B, E, and H are keyed to the second-motion shaft, but are free to slide along it. I and J are keyed upon the third-motion shaft, and all three of the face-plate pinions can be withdrawn. If I is withdrawn with the same mechanism as moves J, pinion H may be fixed to the shaft.

We then have: Single gear, G, F; first double gear, A, B, E, F; second double gear, C, D, E, F; first treble gear, A, B, H, I, J, F; second treble, C, D, H, I, J, F.

In this case also it is necessary to fit a reversing motion upon the countershaft, in order to secure the correct direction of rotation when running in double gear.

Number of spindle speeds $= 5pq$.

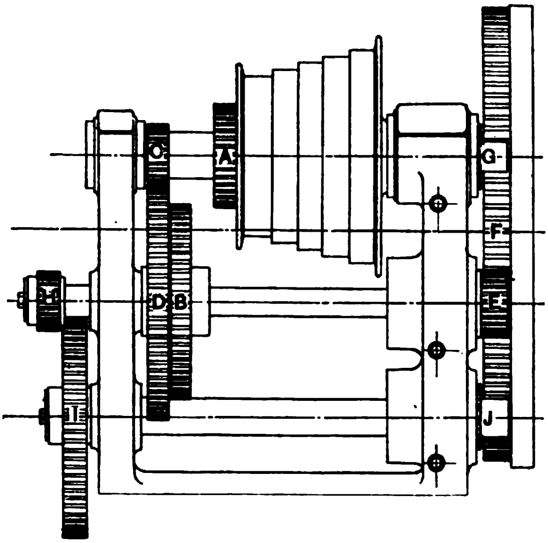


FIG. 54.

The next figure, No. 55, shows an arrangement giving two double and two treble gears, and by which also a high value of R_o is obtained. Here the cone and wheels E, C, J, and A are keyed to the first-motion shaft, F, D, G, and I to the second-motion shaft, and all on this shaft are free to slide, F and D moving together, as only one or the other is used at one time. The wheel H is keyed to the spindle. This wheel may be of a fair size, and it is common continental practice to key it to the spindle as shown. In this country it is generally preferred to attach it to the face-plate, within the internal gear, and the wheel G is then overhung when engaged, and is withdrawn within the bearing in a manner similar to that previously described.

The arrangement shown gives two double gears, C, D, G, H and E, F, G, H; two treble gears, C, D, I, J, A, B, and E, F, I, J, A, B. The gear ratios, with about maximum possible gears, are 12, 24, 48, and 96. Their highest value is still relatively small, and in order to increase it some continental firms mount the cone on another shaft and gear down to the face-plate pinion shaft. It may be observed that, by having the cone and wheels E and C coupled together and free on the shaft, yet capable of being locked to it, and

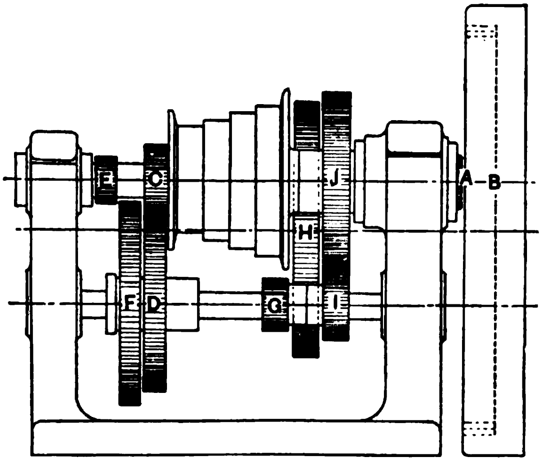


FIG. 55.

by providing space to allow of wheels F and D being disengaged, a single gear in addition to the above can be obtained.

In Fig. 56 is shown an arrangement, similar to Messrs. Richards', capable of giving two single gears and four treble gears. The first single gear is intended for small work and a polishing speed. The cone and wheels A and C revolve together on the side shaft; wheels F and H are keyed to it, F carries a lock bolt which fits the recesses in the cone, and is locked to it when running single gear. The wheels B, D, and E are mounted on a back shaft—B and D being free to slide on same—which has its bearing in two excentric bushes in the headstock cheeks. These bearings are coupled together by means of a handle, and throw the wheels, D, B, and E in or out of gear. First single gear, F, G; second single gear, H, I; first treble gear, A, B, E, F, G; second treble gear, C, D, E, F, G; third treble gear, A, B, E, F, H, I; fourth treble gear, C, D, E, F, H, I. Here the ratio of F to G is comparatively small, and if the speeds are all arranged in geometrical progression, this value of R_e is useless for high-speed and compromise lathes generally; but it is here introduced for polishing and small drilling and boring. It need not be in geometrical progression with the other speeds—*i.e.* there will be a gap in the geometric ratio above the single gear H, I. For all gears of higher ratios the speeds may be in geometrical progression.

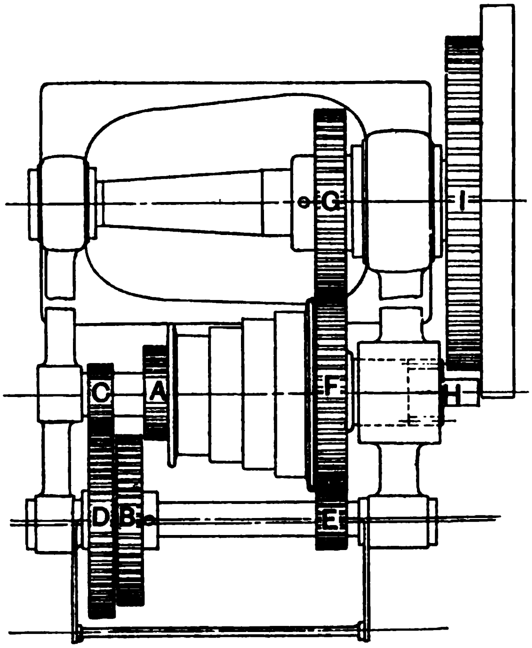


FIG. 56.

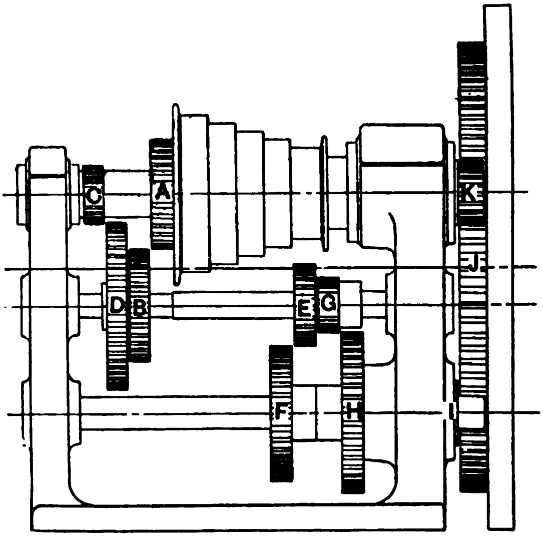


FIG. 57.

Fig. 57 shows an arrangement giving one single and four treble gears. The cone and wheels A, C, K are all keyed to the shaft, K being capable of withdrawing within the bearing. Wheels D, B, E, and G are keyed to the second-motion shaft, and can slide along it, whilst F, H, and I are similarly arranged upon the third-motion shaft. The drives then are: Single gear, K, J; first treble, A, B, E, F, I, J; second treble, A, B, G, H, I, J; third treble, C, D, E, F, I, J; fourth treble, C, D, G, H, I, J. Number of spindle speeds = $5pq$.

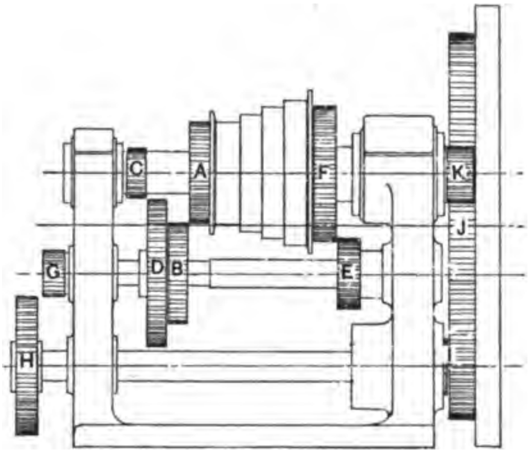


Fig. 58.

Fig. 58 gives another method which has been used for obtaining the same number of back gears, viz. one single and four trebles. The cone and wheels A and C here run together freely upon the first-motion shaft, instead of being keyed to it as in Fig. 57. By then mounting wheel F upon the first-motion shaft and keying the same to it, along with pinion K, the single gear drive is obtained through K and J as before by locking the cone to F. The only other variation from Fig. 57 is that the pinion and wheel G and H are overhung upon the second- and third-motion shafts. Single gear, K, J—cone locked to F; first treble, A, B, E, F, K, J; second treble, A, B, G, H, I, J; third treble, C, D, E, F, K, J; fourth treble, C, D, G, H, I, J.

This arrangement is in several respects inferior to that of Fig. 57. The cone, A and C run loose upon the spindle, instead of being always keyed to it. The overhanging of G and H does not produce any corresponding shortening of the headstock; whilst, on the contrary, the mounting of the wheel F between the cone and the bearing tends to increase its length. The wheel H is a very long way from the face-plate pinion I, and will necessitate the third-motion shaft being made exceptionally stiff. The arrangement of Fig. 58 also necessitates the use of a lock-plate, which is not the case with that of Fig. 57.

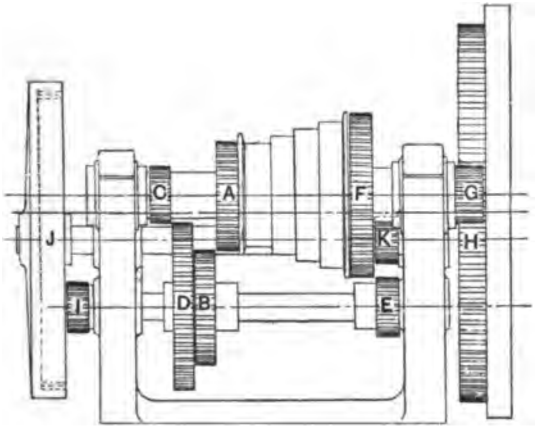


Fig. 59.

We now come to arrangements which give not only double or treble, but quadruple gears also. Fig. 59 shows an arrangement similar to Messrs. Shanks', wherein the third-motion shaft carries an internally geared wheel, whereby one single, two treble, and two quadruple gears are obtained. The cone and pinions A and C are mounted freely upon, and the wheel and pinion F and G are keyed to, the first-motion shaft, F carrying a lock-bolt for the purpose of the single-gear drive. I, D, B, and E are keyed to the second-motion shaft, and all but the first can slide along it. The third-motion shaft—situated between the first- and second-motion shafts, and at the back of the same—carries the internal wheel J and the pinion K, both of which can slide along it. [When F is larger than the cone end plate, K and J may both be withdrawn at one operation.] We thus have: single gear—cone locked to F—G, H; first treble gear, A, B, E, F, G, H; second treble gear, C, D, E, F, G, H; first quadruple gear, A, B, I, J, K, F, G, H; second quadruple gear, C, D, I, J, K, F, G, H. Number of spindle speeds equals $5pq$.

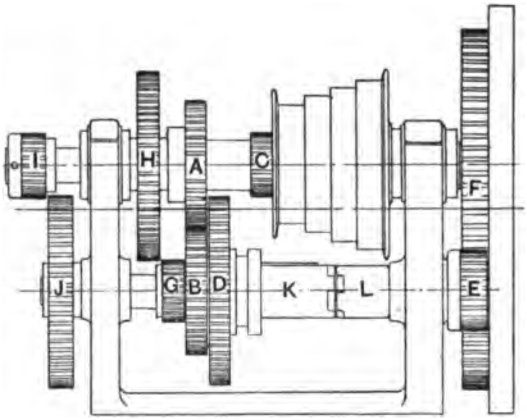


Fig. 60.

Fig. 60 shows an arrangement recently illustrated in *The Engineer* for a 54-in. lathe, by Messrs. Hulse and Co., to give two double and two quadruple gears.

The pinions A and C are fast to the cone, and revolve freely on the shaft, whilst the wheel H and pinion I are keyed to it, the latter being capable of sliding on the same. J, E, and clutch L, are keyed to the second-motion shaft, whilst the clutch K is freely mounted on it. G and K may be in one piece, and the wheels B and D are keyed together, and to the sleeve K, and are capable of sliding on same. When withdrawing B and D the sleeve K is temporarily locked to the shaft by means of a set screw, and the set screw by which B and D are locked to the sleeve being released, a claw fitted with two ferrules, and carried on a rack which is actuated by a hand-wheel and rack pinion, moves B and D in or out of gear. When G is wanted to engage with H, the wheels B and D are withdrawn sufficiently far to enable G to move forward, B and D are then locked to the sleeve by means of the set screw, and the above withdrawing mechanism is used to move G into gear. When running with double gear the pinions I and G are withdrawn, and the clutch K engaged with L. When using the quadruple gear, the clutch K is withdrawn and the pinions G and I engaged with H and J. First double gear, A, B, E, F; second double gear, C, D, E, F; first quadruple gear, A, B, G, H, I, J, E, F; second quadruple gear, C, D, G, H, I, J, E, F.

In order to keep the length of the headstock down, the wheels I and J—Fig. 60—are outside the bearing, and when the lathe is taking its maximum cut, the greatest torque coming on the second-motion shaft and the wheel J being situated at the end of the shaft furthest from the work, requires to be very large in diameter in order to give the necessary stiffness. This can be reduced by bringing the wheel J forward to the front of the headstock, as shown in Fig. 61. As in Fig. 60, the wheels A and C and the cone are fixed together and revolve freely on the shaft, while H and I are keyed to it, I being free to slide on the same. G, B, D and K are keyed to a short shaft on which they are all free to slide. L, J, and E are keyed to a second short shaft on the same axis prolonged as the first-mentioned short shaft. When running double gear the clutch K is moved forward to engage with L; G and I being withdrawn. When running quadruple gear, G and I are engaged, and the clutch K is disengaged from L. First double gear, A, B, E, F; second double gear, C, D, E, F; first quadruple gear, A, B, G, H, I, J, E, F; second quadruple gear, C, D, G, H, I, J, E, F. Some makers object to the short lengths of shaft, as the vertical reaction on the face-plate pinion tends to make the second short shaft bearing bell-mouthed.

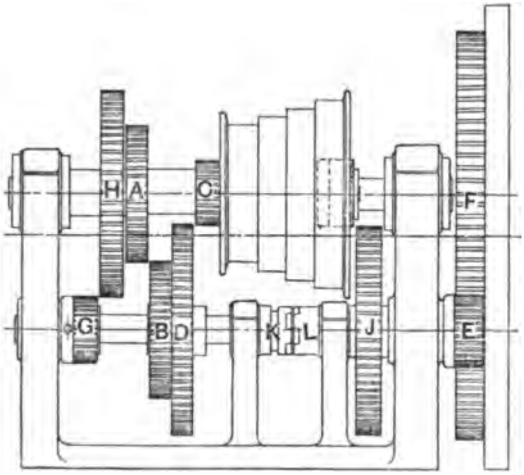


Fig. 61.

Fig. 62 shows a modification of 60 and 61, which also dispenses with the short shafts and long sleeves, the heaviest wheel being close to the face-plate pinion. The cone and wheels A and C may be keyed to the shaft, so that the wear due to them takes place in the bearings where it can be taken up. The wheel H is keyed to the sleeve of the pinion I, and revolves freely on the shaft. It will be clear that the wear due to H, I is considerably smaller than that due to the cone, as the rubbing speed is only that due to the difference between the cone shaft and gear revolutions, and this wear takes place when using the quadruple gear only. The wheel J is keyed to the sleeve of the pinion E, which is free on the second-motion shaft, while B, D, and G are keyed to and are capable of sliding along it. When using the double-gear pinion G, which has a clutch face, is made to engage with the clutch face on the wheel J or pinion sleeve E, the wheels H and I being withdrawn, and conversely, when using the quadruple gear. We then have the following drives: first double gear, A, B, E, F; second double gear, C, D, E, F; first quadruple gear, A, B, G, H, I, J, E, F; second quadruple gear, C, D, G, H, I, J, E, F.

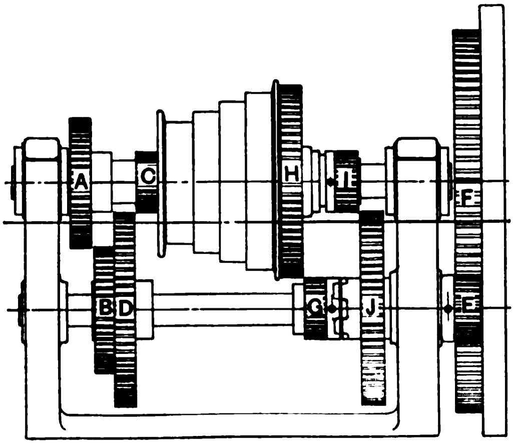


Fig. 62.

It is evident that the arrangement shown on Fig. 62 could be made to give three double and two quadruple gears by locking the wheel H with the cone and engaging I with J. This arrangement does not, however, give such a large ratio of maximum to minimum speeds in geometrical progression as is possible with the other. This arises from the necessity of diminishing the ratio of the two gears H, G, and D, C; which must be done to secure a proper geometrical progression of speeds with the second arrangement.

Fig. 63 shows a simple arrangement known as the Whitworth principle, where the cone and all the wheels are keyed to the shafts, and clutches and sleeves dispensed with.

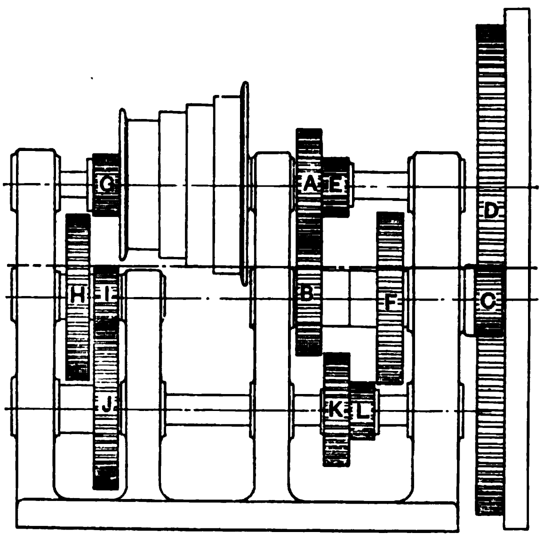


FIG. 63.

The second-motion shaft is in two pieces, and the arrangement permits of two double gears and two quadruple gears, giving a very high maximum ratio of gear. We have: first double gear, A, B, C, D; second double gear, E, F, C, D; first quadruple gear, G, H, I, J, K, B, C, D; second quadruple gear, G, H, I, J, L, F, C, D.

Fig. 64 shows an arrangement where the cone is brought to the rear of the head, three double and four quadruple gears being obtained. The cone is keyed to the shaft carrying the pinions A, C, and E. A second-motion shaft situated behind and above the cone shaft carries the wheels B, F, J, and G; whilst another second motion shaft below and behind the cone shaft carries the wheels D, M, and H. Behind and between these two second motion shafts a third shaft is placed, upon which wheels K, L, and N are mounted. The drives then are: first double gear, A, B, G, I; second double gear, C, D, H, I; third double gear, E, F, G, I; first quadruple gear, A, B, J, K, N, D, H, I; second quadruple gear, A, B, J, K, L, M, H, I; third quadruple gear, E, F, J, K, N, D, H, I; fourth quadruple gear, E, F, J, K, L, M, H, I.

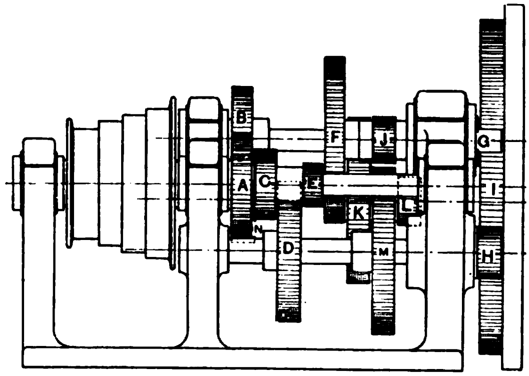


FIG. 64.

Fig. 65 shows an arrangement by a continental firm where the cone and wheel A are keyed to a side shaft; the drive to the spindle being obtained through two or more pairs of wheels, thus giving a high value of R°, the ratio of cone to spindle speed when the latter has its highest speed of rotation (N_g). The first-motion shaft, which is immediately behind the cone shaft, carries the wheels B, L, C, I, and J. On a second-motion shaft behind and above the first-motion shaft are keyed the wheels E and D, while another second-motion shaft below the former carries the wheels M, G, and H. On the spindle is keyed the wheel F. B and L are coupled together and rotate freely on the shaft, while C is keyed to it, and has a clutch face which engages with a corresponding face on L. M, G and H are keyed to their shaft, but free to slide on same. Thus the arrangement is capable of giving one double gear, one treble gear, three quadruple, and one quintuple gear. Thus: double, A, B, J, K; treble, A, B, C, D, E, F; first quadruple: A, B, C, G, H, I, J, K; second quadruple: A, B, L, M, G, C, D, E, F or A, B, L, M, G, C, J, K; third quadruple: A, B, L, M, H, I, J, K; quintuple, A, B, L, M, H, I, C, D, E, F.

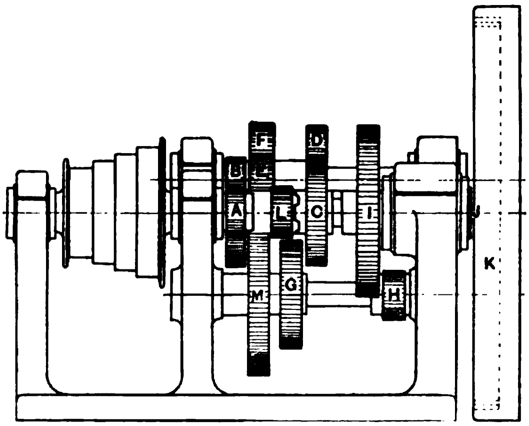


FIG. 65.

Returning now for a moment to Figs. 57 and 58, we observe that by adding two more pairs of wheels which are placed upon either the second- and third-motion shafts, as in Fig. 66, or upon the second- and first-motion shafts, as in Fig. 67, we may obtain four quintuple gears in addition to the single gear, and four treble gears. Referring to Fig. 66, it will be seen that, instead of employing only three shafts, a fourth has been inserted. It is perfectly possible to carry out the arrangement as above described, but the design we have figured—66—is preferable, as the length of the headstock is not increased over that of Fig. 57, and the wheels O and N are always keyed in position, instead of being loose and sliding on the shaft.

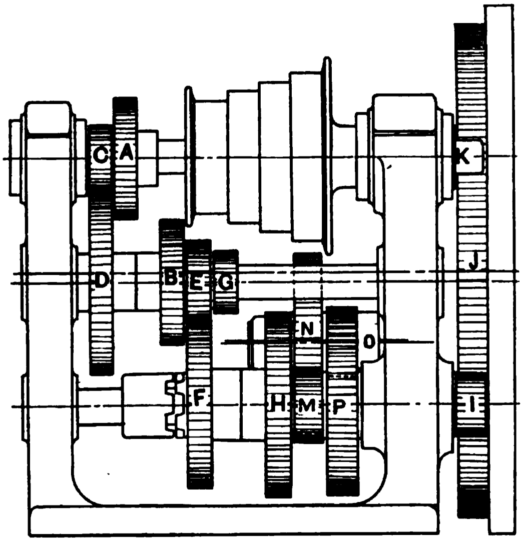


FIG. 66.

The cone and wheels A, B, C, D, E, G, and K are arranged exactly as previously described for Fig. 57. On the third-motion shaft, however, F, H, and M are keyed together upon a sleeve, which can be clutched to it or not as desired. An additional wheel P also is permanently keyed upon this shaft. The wheels N and O, situated between and behind the second- and third-motion shafts, and mounted upon a short fourth shaft, mate with M and P. The drives then are: single gear, K, J; first treble, A, B, E, F, I, J;

second treble, A, B, G, H, I, J; third treble, C, D, E, F, I, J; fourth treble, C, D, G, H, I, J; first quintuple, A, B, E, F, M, N, O, P, I, J; second quintuple, A, B, G, H, M, N, O, P, I, J; third quintuple, C, D, E, F, M, N, O, P, I, J; fourth quintuple, C, D, G, H, M, N, O, P, I, J. The arrangement of Fig. 67 is similarly derived from Fig. 58, and the drives are: single gear, K, J; first treble, A, B, O, P, K, J; second treble, A, B, E, F, K, J; third treble, C, D, O, P, K, J; fourth treble, C, D, E, F, K, J; first quintuple, A, B, G, H, M, N, O, P, K, J; second quintuple, A, B, G, H, M, N, E, F, K, J; third quintuple, C, D, G, H, M, N, O, P, K, J; fourth quintuple, C, D, G, H, M, N, E, F, K, J. As already remarked in connection with Figs. 57 and 58, we consider this arrangement in many ways inferior to that of Fig. 66.

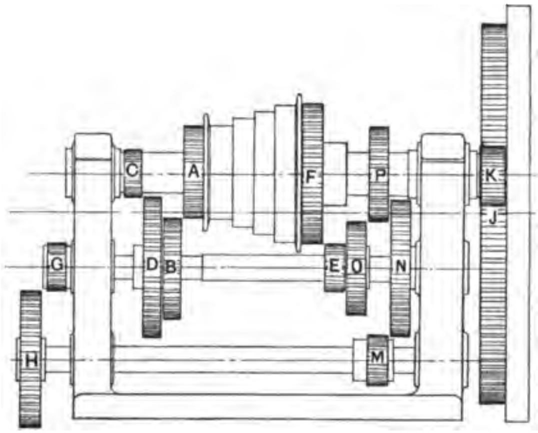


Fig. 67.

In addition to the defects specified: the overhanging of G, H without any corresponding shortening of the head, the great distance between H and M, and the provision of only one face-plate pinion; it is clearly bad practice to employ the gears O, P and E, F, which must of necessity have coarsely pitched teeth to transmit the heavy drive of the quintuple gear, also for the light loads and high speeds of the treble drives. The corresponding gears of Fig. 66, viz. M, N and O, P, which are heavy and of large diameter, are only used for quintuple driving.

CHAPTER XXI

THE FAST HEADSTOCK (*continued*)—GEAR ARRANGEMENTS

ALL-GEAR HEADSTOCKS

IN the types of headstock described in our last two chapters, variation of speed was obtained by the use of gear wheels and a stepped cone, the latter being mounted either upon the spindle or upon a back shaft. We now propose to describe a few of the most recent examples of headstocks in which the cone is displaced by a single pulley driven by a belt, or by a spur or sprocket wheel driven from an electric motor.

When driving from a main shaft, some makers prefer to arrange for making the required close variations of speed by gearing or pulleys overhead —*i.e.* between the main and countershafts — the coarser changes being obtained on the headstock itself. What advantages this method has over that in which all the gears are arranged within the head itself we leave for subsequent discussion. Similarly, with motor-driven lathes the speed variation may be obtained partly in the motor and partly by gear in the head. As a general rule, the motor can give the same variation of speed ratio as the displaced cone, but continuously, instead of *per saltum.* In some cases, however, the whole range of variation is obtained in the motor itself. The relative advantages of these two methods will also form the subject of future inquiry.

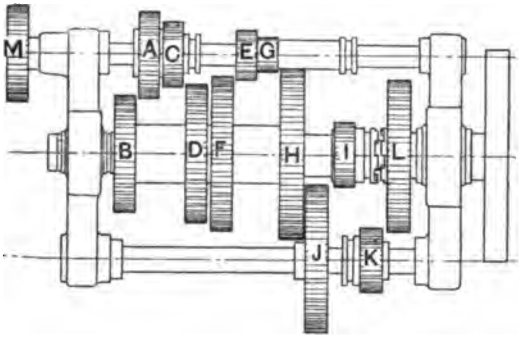


FIG. 68.

It is evident that the substitution of a belt pulley, a spur or a sprocket wheel for the stepped cone of Chapters XIX. and XX. will convert any of the arrangements there described into a so-called "all-gear head"; but that there must be a suitable speed-varying device behind this driving wheel, *e.g.* an electric motor to fill in the gap caused by the removal of the cone.

Fig. 68 shows an arrangement by J. Buckton and Co., Limited, Leeds, which is fully described in the "Machine Tool Supplement" of *The Engineer* of November 4, 1904. Here the wheels M, A, C, E, G are keyed to a side shaft at the back of the headstock; the wheel M meshes with a pinion on the end of the armature shaft of the motor, while the pinions A, C, and E, G, slide in pairs on the shaft, and respectively mesh with the wheels B, D, and F, H, keyed to a sleeve revolving freely on the spindle. A second side shaft mounted at the front of the headstock carries the wheels J and K, the latter being free to slide on the shaft. The pinion I is also keyed to the spindle sleeve, and has a clutch face. When running single gear it meshes with a corresponding face on the wheel L, which is keyed to the spindle. The arrangement thus permits of four single gears and four treble gears.

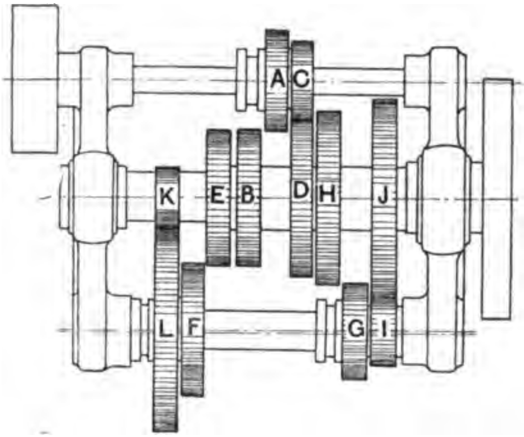


Fig. 69.

Single gears: I engaged with L; K withdrawn; first, A, B; second, C, D; third, E, F; fourth, G, H.

Treble gears: I withdrawn from L and engaged with J; K engaged with L; first, A, B, I, J, K, L; second, C, D, I, J, K, L; third, E, F, I, J, K, L; fourth, G, H, I, J, K, L.

By transferring two of the wheels from the first to the third-motion shaft of Fig. 68, an arrangement similar to Fig. 69 is obtained. This will give two single and eight treble gears; or ten speeds as against eight in the former case. The arrangement shown in Fig. 69 is after the patent of the Draper Machine Tool Company, Worcester, Mass. Two pinions A and C are coupled together and slide on the back shaft. The wheels K, E, B, and D are keyed to a sleeve revolving freely on the spindle, the last wheel being provided with recesses to receive the lock bolt carried by the wheel H when running single gear. H and J are keyed to the spindle, while the wheels L, F, and G, I, are keyed to a sleeve running freely on an excentric shaft, and slide in pairs on the same.

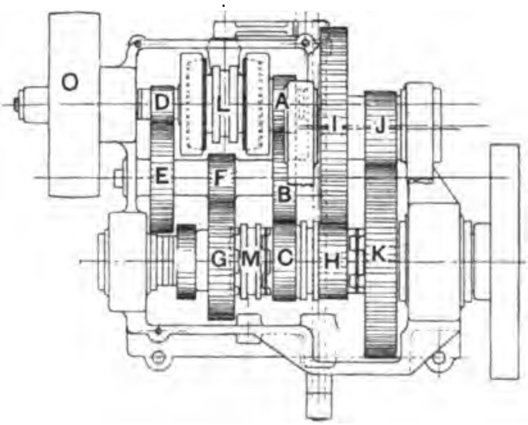


Fig. 70.

When running single gear the wheel D is locked to H, the wheels L, F, G, I being thrown out of gear.

Single gears: first, A, B; second, C, D.

Treble gears: first, A, B, E, F, G, H; second, C, D, E, F, G, H; third, A, B, E, F, I, J; fourth, C, D, E, F, I, J; fifth, A, B, K, L, G, H; sixth, C, D, K, L, G, H; seventh, A, B, K, L, I, J; eighth, C, D, K, L, I, J.

Fig. 70 shows an arrangement by the Tangye Tool and Electric Company, which was illustrated and described in the "Machine Tool Supplement" of *The Engineer* of November 4, 1904. It is capable of giving one single, three double, one treble, and three quadruple gears.

There are two side shafts carried at the back of the headstock, on one of which is keyed the driving pulley O and friction clutch L, while the pinions D and A revolve freely on the same. On the second side shaft are keyed the wheels E, F, and B. Running freely upon a loose sleeve mounted independently on the spindle are two wheels G and C, which have clutch faces to mesh with clutch M. This clutch and the pinion H, which also has a clutch face, are keyed to the sleeve. The wheel K may be clutched to the pinion H. and is keyed to the spindle. Vertically above the spindle is an excentrically mounted shaft carrying upon a sleeve the two wheels I and J. The four wheels H, I, J, and K form an arrangement similar to the double back gear in small lathes. For convenience this sleeve with wheels and bearings is here shown rebated backwards. The shafts and spindles rotate in self-oiling bearings, and the whole of the gear is enclosed in a casing, all changes being effected by levers placed outside the same. When running single gear the clutch L is engaged with A, the gears I and J disengaged, M is clutched to C, and H to K.

Single gear: A, B, C.
Double gears: First, D, E, B, C (L engaged with D, M with C).
Second, A, B, F, G (L „ A, M „ G).
Third, D, E, F, G (L „ D, M „ G).
(H clutched to K throughout, I and J withdrawn.)

In the treble and quadruple gears the pinion H is withdrawn from K, and the wheel and pinion I and J engaged. The clutch positions for the treble gear are the same as for the single gear. For the quadruple gears the clutches are similarly placed as for the corresponding double gears, the only addition being the inclusion of the wheels I and J.

Treble gear: A, B, C, H, I, J, K (L engaged with A, M with C).
Quadruple gears:
First, D, E, B, C, H, I, J, K (L engaged with D, M with C).
Second, A, B, F, G, H, I, J, K (L „ A, M „ G).
Third, D, E, F, G, H, I, J, K (L „ D, M „ G).

Being an all-gear head, this arrangement is capable of taking a heavy cut; yet the Tangye Company do not recommend this principle of headstock for lathes above 12-in. centres. It will be observed that the high speed of pulley shaft permits of a friction clutch being used, although a large amount of power has to be transmitted.

The arrangement shown in Fig. 71 as recently made by Hulse and Co., Limited, Manchester, permits of two single, two double, two treble, and two quadruple gears being obtained. The sprocket wheel M is keyed to the sleeve of the pinion I, which revolves freely on the shaft. The wheel M, driven by a chain and pinion at the end of the armature shaft, may, of course, be replaced by a pulley. The wheels L, A, C, and clutch N are keyed to the first-motion shaft, whilst the wheels E, F, and pinion G are keyed to the second-motion shaft; J is keyed to the sleeve of the pinion K, and revolves freely on the shaft. The wheels B and D are keyed to the spindle.

Single gears:	First,	A, B,	clutch N engaged with I.
	Second,	C, D,	„ „
Double gears:	First,	A, E, G, H,	„ „
	Second,	C, F, G, H,	„ „
Treble gears:	First,	I, J, K, L, A, B,	N withdrawn.
	Second,	I, J, K, L, C, D,	„
Quadruple gears:	First,	I, J, K, L, A, E, G, H,	N withdrawn.
	Second,	I, J, K, L, C, F, G, H,	N „

Fig. 72 shows an arrangement by Sharp, Stewart and Co., where the motor is mounted on the headstock, the drive to the first-motion shaft being obtained through the sprocket pinion and wheel A and B.

The first-motion shaft, situated above and behind the face-plate pinion shaft, carries the wheels C and D, which are keyed to but free to slide upon it. On the face-plate pinion shaft is mounted a sleeve, which is free to rotate independently of it, and capable of being locked to it by means of a lock plate P and the lock bolt carried by the wheel L. The wheels E, F, G, I and lock plate P are keyed to the sleeve, while the wheel L and pinion M are keyed to the shaft.

On the quadruple gear shaft are keyed the wheels J, H, and K, all of which are free to slide. The arrangement is identical with that shown in Fig. 53, except that the cone has been displaced by the wheels E and F, and the first-motion shaft has been added to give two reductions of speed from the motor.

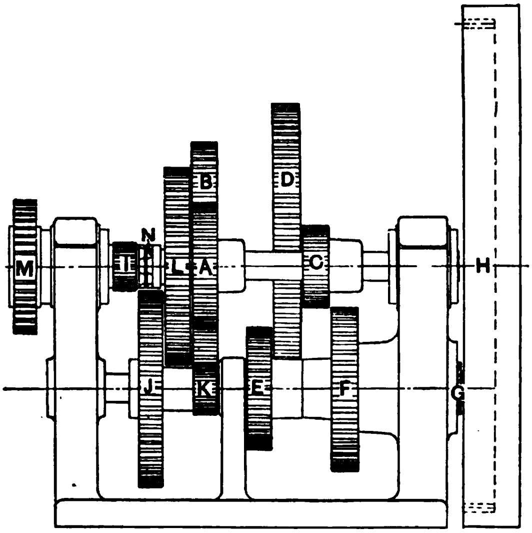


Fig. 71.

The arrangement thus permits of two double and four quadruple gears—neglecting the wheels A and B.

Double gears: First, A, B, C, E, M, N; second, A, B, D, F, M, N.

Quadruple gears: First, A, B, C, E, G, H, K, L, M, N; second, A, B, D, F, G, H, K, L, M, N; third, A, B, C, E, I, J, K, L, M, N; fourth, A, B, D, F, I, J, K, L, M, N.

Fig. 73 shows an arrangement patented by the late Mr. W. W. Hulse in January, 1896, which is similar to that shown in Fig. 72, except that any number of wheels may be keyed to the sleeve, all capable of gearing with a pinion, the distance from the centre of whose shaft to that of the sleeve is capable of variation, so that the pinion can engage with any one of the series of wheels upon the sleeve. Several methods of effecting this change of centres are covered in the specification; as, *e.g.*, by a pair of swinging arms, by a compound slide, by slide and arms combined, etc. The drive to the pinion shaft from the motor may be through a worm, worm wheel, and spur gearing, when the motor is at right angles to the lathe centre line—as shown in the sketch. Other suitable power-transmitting arrangements are also indicated. Several arrangements similar to this have recently appeared, but the modifications seem to be more in details than in principle. In the "Machine Tool Supplement" of *The Engineer* of November 4, 1904, there is shown a detailed drawing of a recent improved form of this patent by Hulse and Co., Limited.

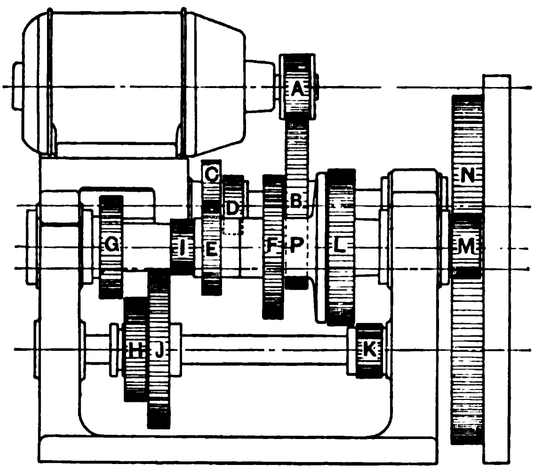


FIG. 72.

The arrangement shown in Fig. 73 carries a worm wheel A and spur wheel B coupled together and revolving freely on the shaft. Motion is communicated from the motor through a worm which drives A and B; the latter in turn drives the wheel C keyed to the end of jockey-pinion shaft, which drives the wheel D. By swinging the arms and sliding the wheel D along the shaft, it can be made to mate with any one of the series E, F, G, H. On the first-motion shaft is keyed the wheels N and face-plate pinion O, and a sleeve riding freely on this shaft has keyed to it the wheels E, F, G, H, I, K, and a lock plate. On the second-motion shaft are keyed the wheels J, K, and M, the last being free to slide. The worm and worm wheel A and spur wheels B and C may be considered as speed-reducing gears, and are common to all the gears; the arrangement then is capable of giving four double and eight quadruple gears.

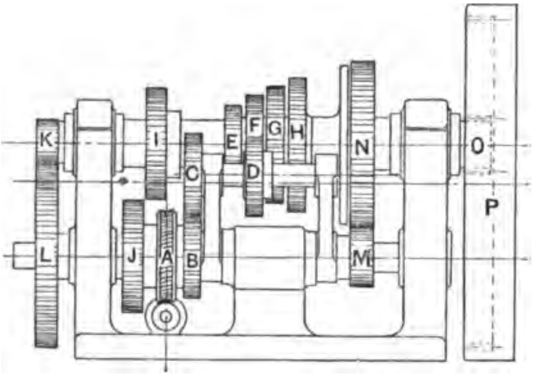


FIG. 73.

Double gears: first, D, E, O, P; second, D, F, O, P; third, D, G, O, P; fourth, D, H, O, P.

Quadruple gears: first, D, E, I, J, M, N, O, P; second, D, F, I, J, M, N, O, P; third, D, G, I, J, M, N, O, P; fourth, D, H, I, J, M, N, O, P; fifth, D, E, K, L, M, N, O, P; sixth, D, F, K, L, M, N, O, P; seventh, D, G, K, L, M, N, O, P; eighth, D, H, K, L, M, N, O, P.

Although the arrangement shown in Fig. 71 is capable of giving in all eight speeds—two single, two double, two treble, and two quadruple—the range of the ratio is comparatively small. The large wheel on the spindle is limited in diameter, and is consequently not so suitable for large lathes as that shown in Fig. 74, which gives two double and four quadruple gears. A detailed drawing of a headstock with an arrangement of gear similar to this by Hulse and Co. is shown in the "Machine Tool Supplement" of *The Engineer* of November 4, 1904.

In Fig. 74 the pulley and wheels A and C are keyed to a sleeve free on the first-motion shaft, whilst H, F and I are keyed to the shaft, the last two being free to slide together on the same. On the second-motion shaft are keyed the wheels D, B, G and E; the last fitted with a lock-plate. J is keyed to the sleeve of the face-plate pinion K, which revolves freely on the shaft, but is capable of being locked to it by means of a lock bolt carried by J and the lock-plate. The sets are as follows:—

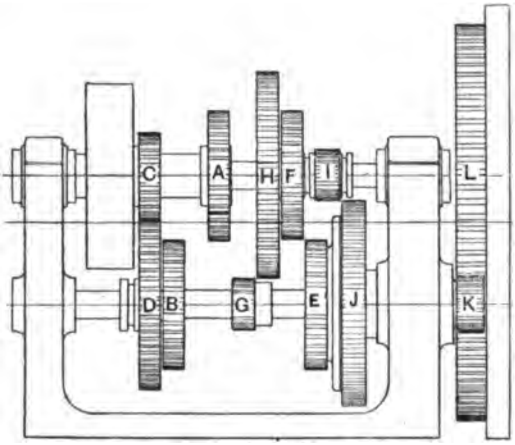


FIG. 74.

Double gears: First, A, B, K, L; second, C, D, K, L.

Quadruple gears: First, A, B, E, F, I, J, K, L; second, C, D, E, F, I, J, K, L; third, A, B, G, H, I, J, K, L; fourth, C, D, G, H, I, J, K, L.

Fig. 75 shows an arrangement similar to that shown in Fig. 70, by means of which sixteen instead of eight speeds are obtained by the addition of three wheels. The driving pulley and friction clutch O are keyed to a side shaft, which also carries the pinions A and D free on the same, but capable of being coupled to it by means of the clutch, which is actuated by a lever at the front of the headstock. The wheels B, G, E and I are keyed to an intermediate shaft. A sleeve, mounted freely upon the spindle, has keyed to it the clutches P and R and the pinion K, whilst the wheels C, H, F and J, having clutch faces, are free, but can be locked to the sleeve by the clutches. The wheel N is keyed to the spindle and has a clutch face corresponding to that of the pinion K, whilst L and M are keyed to a sleeve, free upon an excentric shaft which engages or disengages these gears. The arrangement of the gears are as follows:—

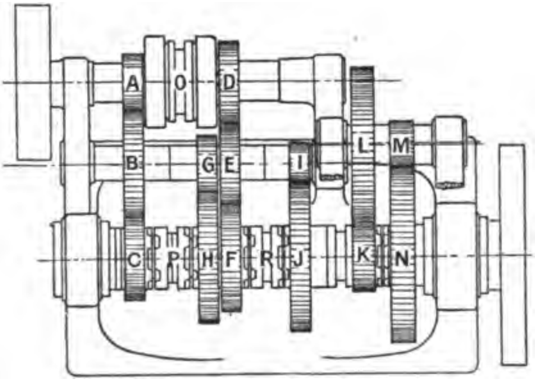


FIG. 75.

Single gears: First, A, B, C; second, D, E, F.

Double gears: First, D, E, B, C; second, A, B, E, F; third, D, E, G, H; fourth, A, B, G, H; fifth, D, E, I, J; sixth, A, B, I, J.

Treble gears: First, A, B, C, K, L, M, N; second, D, E, F, K, L, M, N.

Quadruple gears: First, D, E, B, C, K, L, M, N; second, A, B, E, F, K, L, M, N; third, D, E, G, H, K, L, M, N; fourth, A, B, G, H, K, L, M, N; fifth, D, E, I, J, K, L, M, N; sixth, A, B, I, J, K, L, M, N.

It will be observed that some of the gears other than the single give a ratio that can easily be obtained by a single gear, and to this extent should possibly not be classed as double, treble, or quadruple gears. For the first eight speeds the wheels L and M are disengaged, and the clutch pinion K made to engage with N. The pinion K and clutches P and R are actuated by levers on the front of the headstock casing; provision being made on these levers to prevent more than one of these clutches being engaged at any one time, whilst the shoulder on the pinion K prevents L and M mating with K and N when K is engaged with N.

As a rule, in all gear headstocks the wheels are lubricated either by running in an oil bath or by trailing wicks; some such lubricating device being necessary where steel gears are used.

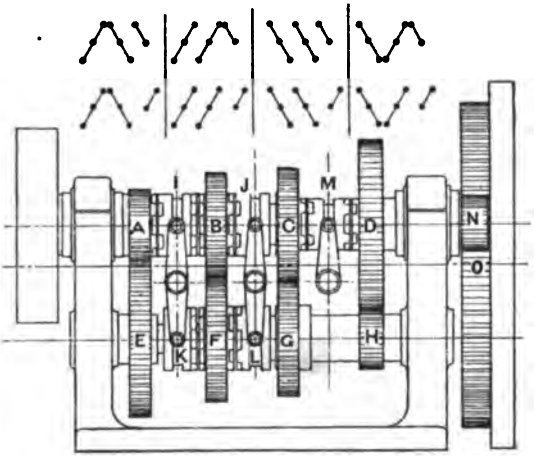


Fig. 76.

It nearly always happens in multiple-gear arrangements that more than one set of gears may be positively in gear at one time; which would, of course, lead to stoppage or breakage. This is particularly the case where sliding clutches are employed.

The arrangement shown in Fig. 76 — after Ruppert, v. *Zeitschrift des Vereins Deutscher Ingenieure,* 1904, page 418—is absolutely interlocking in itself; no matter what positions the levers occupy, there can be no stoppage or breakdown.

The driving pulley and clutch I are keyed to a sleeve revolving freely on the shaft, whilst the wheels A and B revolve freely on the sleeve. The wheel A has a single-clutch face, and B a double-clutch face. The clutch J is keyed to the sleeve of the wheel C, which revolves freely on the shaft; but a clutch face on the right-hand boss of C permits it being coupled to the shaft by a clutch M. The wheel D, which has a clutch face on the left boss, is also loose on the shaft, but can be coupled to it by the clutch M; the face-plate pinion N is keyed to the shaft. On the second-motion shaft are keyed the wheel E and single-faced clutch K. The wheels G and H and single-faced clutch L are keyed to a sleeve revolving freely on the second-motion shaft, while the wheel F is free on the sleeve, and has a double-clutch face, which permits of it being locked to the shaft, or sleeve, or both.

The arrangement of the gears is as follows:—

First, N, O (I engaged with B, J with B, and M with C).

Second, B, F, G, C, N, O (I engaged with B, L with F, and M with C).

Third, A, E, F, B, N, O (I engaged with A, K with F, J with B, and M with C).

Fourth, A, E, G, C, N, O (I engaged with A, K and L with F, and M with C).

Fifth, C, G, H, D, N, O (I engaged with B, J with B, and M with D).

Sixth, B, F, H, D, N, O (I engaged with B, L with F, and M with D).

Seventh, A, E, F, B, C, G, H, D, N, O (I engaged with A, K with F, J with B, and M with D).

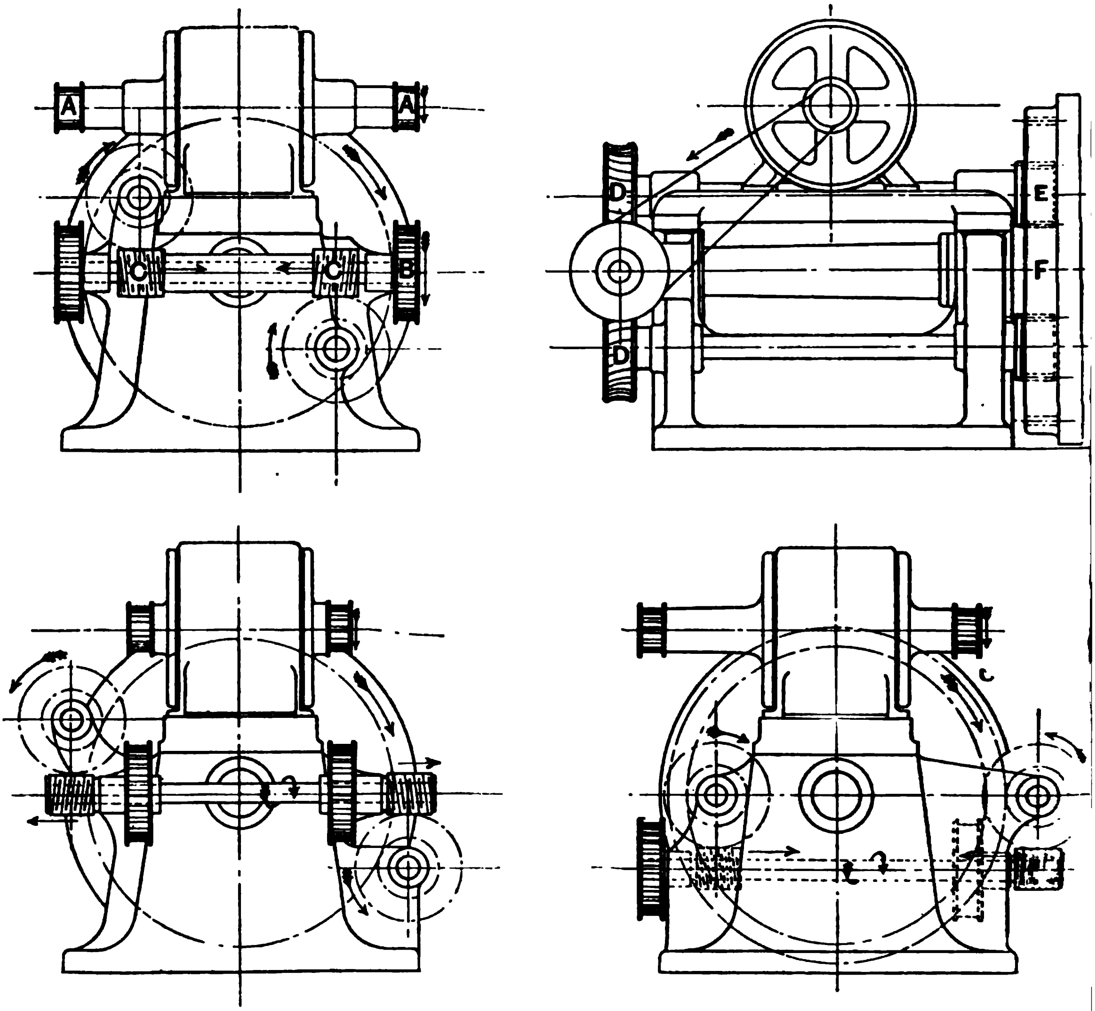


Fig. 77.

Eighth, A, E, H, D, N, O (I engaged with A, K and L with F, and M with D).

The positions in order of the levers for each of these gears are shown immediately over the figure, and read from left to right.

Fig. 77 shows a recently patented device by Hulse and Co., Limited, where the speed variation is obtained wholly in the motor, or in some suitable variable speed device, which is independent of the headstock.

In large lathes, where the cuts are heavy, and frequently intermittent, the load on the face-plate pinion is very great, and there is a considerable amount of chatter, due to the irregularity of the work, the cutting action of the tool at slow speeds, and the large pitch of the pinion tooth necessary for strength. To overcome in part these defects, the arrangement shown (Fig. 77) has two face-plate pinions, and two reversely threaded worms, which drive corresponding worm wheels keyed to the two face-plate pinion shafts.

The worms are keyed to a shaft, or to a sleeve, free on the same, and provision is made to allow the shaft or sleeve to move endwise, so that the thrust of the worm is self-contained therein, and is not received by the bearings.

This floating movement also allows the worms to adjust themselves to suit any inequalities there may be in the gears. By this means the load is divided among the gears, so as to stress them all equally. The directions of the drives and the thrusts of the worms are shown in the figure by the arrows, which may convey a clearer idea of the device. The arrangement is equally suitable for either external, internal, or combined external and face-plate wheels.

It will be observed that there is a pinion at each end of the armature shaft, thus dividing the torque, and permitting of a more rigid drive. By means of a chain the pinions A drive the wheels B, which are keyed to the worm shaft; the worms C drive the wheels D, which are keyed to the end of the face-plate pinion shaft, and E drives F.

CHAPTER XXII

SPINDLE-SPEED RANGE—GEAR AND GEOMETRIC RATIOS WITH DIFFERENT DESIGNS OF HEADSTOCK

It is our intention in the present chapter to enter upon a discussion of the general principles governing the choice of, and designs for, the various types of headstocks described and illustrated in Chapters XIX., XX., and XXI. To this end we shall first put down in tabular form the relations which obtain between the geometric ratio, the number of cone steps, and the gear ratios for one, two, or any number of countershaft speeds. We shall then, by working out specific cases and by inductive methods resulting therefrom, ascertain the suitability of double, treble, quadruple, etc., arrangements of "back"—or compound—gear for the various sizes and types of lathe, as influenced by the greatest and least number of spindle revolutions, and by the number of speeds it is proposed to supply.

Reverting to Chapter VII., the table of relations given on the opposite page may easily be obtained from that given on pp. 40 to 43.

We may now consider a few of the simpler designs of headstock, beginning with Figs. 38, 39, and 40 of Chapter XIX.; calculating the various gear ratios and the possible ratio of spindle speeds.

Case 1.—One countershaft speed, or $p = 1$. Thus if $s = pqm = 8$; or there are eight spindle speeds. Then Table A, which we use for this case, will read (if $q = 4$, and $m = 2$)—

m	Name of Gear.	Speeds No.	Gear Ratios.
1	Single gear	1 to 4	$\frac{\text{Largest}}{\text{Smallest}}$ cone step $= r^{\frac{q-1}{1}} = r^{\frac{3}{1}}$
2	Double gear	5 to 8	$R_1 = \frac{D}{C} \times \frac{B}{A} = r^q = r^4$

Now let $\frac{B}{A} = g_1$; $\frac{D}{C} = g_2$.

Then $g_1 \times g_2 = r^4$

and if $g_1 = g_2 = g$, say

we have $g^2 = r^4$, or $g = r^2$, or $r = g^{\frac{1}{2}}$

Thus we know r, if we assume a constructionally convenient value for g.

TABLE L

Tables of Values of Ratios of Compound Gears for Headstocks in Terms of the Geometric Ratio and Number of Cone Steps, Gears, and Countershaft Speeds.

A.—There is but one countershaft speed $R_o \genfrac{}{}{0pt}{}{= 1 \text{ if cone is on the spindle.}}{> 1 \text{ if cone is off the spindle.}}$

Number of ways cone can be coupled to spindle, and name of gear.		Speed number.	Value of gear ratio r = geometric ratio.
1	Single gear	1 to q	$\frac{A}{E} = r^{\frac{q-1}{2}}$ $= \frac{\text{largest}}{\text{smallest}}$ cone step
2	First compound gear	$q + 1$ to $2q$	$R_1 = r^{q}$
3	Second compound gear	$2q + 1$ to $3q$	$R_2 = r^{2q}$
m	$(m - 1)$th compound gear . . .	$(m - 1)q + 1$ to mq	$R_{m-1} = r^{(m-1)q}$

B.—There are two countershaft speeds $R_o \overset{=}{>} 1$.

1	Single gear (fast)	1 to q	$\frac{A}{E} = r^{\frac{q-1}{2}}$
	„ (slow)	$q + 1$ to $2q$	$R_c = r^{q}$
2	First compound gear (fast) . . .	$2q + 1$ to $3q$	$R_1 = r^{2q}$
	„ „ (slow) . . .	$3q + 1$ to $4q$	$R_cR_1 = r^{3q}$
3	Second compound gear (fast) . .	$4q + 1$ to $5q$	$R_2 = r^{4q}$
	„ „ (slow) . .	$5q + 1$ to $6q$	$R_cR_2 = r^{5q}$
m	$(m - 1)$th compound gear (fast) .	to $2(m - 1)q$	$R_{m-1} = r^{(2m-2)q}$
	„ „ „ (slow) .	$2(m - 1)q + 1$ to $2mq$	$R_cR_{m-1} = r^{(2m-1)q}$

C.—There are three countershaft speeds $R_o \overset{=}{>} 1$.

1	Single gear (fast)	1 to q	$\frac{A}{E} = r^{\frac{q-1}{2}}$
	„ (medium)	$q + 1$ to $2q$	$R_{2c} = r^{q}$
	„ (slow)	$2q + 1$ to $3q$	$R_{3c} = r^{2q}$
2	First compound gear (fast) . . .	$3q + 1$ to $4q$	$R_1 = r^{3q}$
	„ „ (medium) .	$4q + 1$ to $5q$	$R_{2c}R_1 = r^{4q}$
	„ „ (slow) . . .	$5q + 1$ to $6q$	$R_{3c}R_1 = r^{5q}$
m	$(m - 1)$th compound gear (fast) .	to $(3m - 2)q$	$R_{m-1} = r^{(3m-3)q}$
	„ „ „ (medium)	$3(m - 2)q + 1$ to $(3m - 1)q$	$R_{2c}R_{m-1} = r^{(3m-2)q}$
	„ „ „ (slow) .	$3(m - 1)q + 1$ to $3mq$	$R_{3c}R_{m-1} = r^{3(m-1)q}$

D.—There are any number, p, of countershaft speeds $R_o \overset{=}{>} 1$.

1	Single gear (fastest)	1 to q	$\frac{A}{E} = r^{\frac{q-1}{2}}$
1	„ „	$q + 1$ to $2q$	$R_{2c} = r^{q}$
1	„ „	$2q + 1$ to $3q$	$R_{3c} = r^{2q}$
1	„ (slowest)	$(p - 1)q + 1$ to pq	$R_{pc} = r^{(p-1)q}$
2	First compound gear (fastest) . .	$pq + 1$ to $(p + 1)q$	$R_1 = r^{pq}$
2	„ „ „ . .	$(p + 1)q + 1$ to $(p + 2)q$	$R_{2c}R_1 = r^{(p+1)q}$
2	„ „ „ . .	$(p + 2)q + 1$ to $(p + 3)q$	$R_{3c}R_1 = r^{(p+2)q}$
2	„ „ (slowest) .	$(2p - 1)q + 1$ to $2pq$	$R_{pc}R_1 = r^{(2p-1)q}$
m	$(m - 1)$th compound gear (fastest)	$(m - 1)pq + 1$ to $[pm - (p - 1)q]$	$R_{m-1} = r^{(pm-p)q} = r^{p(m-1)q}$
m	„ „ „ „ „	$[pm-(p-1)q]+1$ to $[pm-(p-2)q]$	$R_{2c}R_{m-1} = r^{\{pm-(p-1)\}q}$
m	„ „ „ „ „	$[pm-(p-2)q]+1$ to $[pm-(p-3)q]$	$R_{3c}R_{m-1} = r^{\{pm-(p-2)\}q}$
m	„ „ „ (slowest)	$(mp - 1)q + 1$ to mpq	$R_{pc}R_{m-1} = r^{(pm-1)q} = r^{s-q}$

Table D is general and converts into any of the others by putting in the proper value of p.

In all the tables p = number of countershaft speeds, m = number of ways cone and spindle can be coupled, and q = number of cone steps.

Now $$S_r = \frac{N_g}{N_l} = r^{8-1} = r^7$$

But $$r = g^{\frac{1}{2}}$$

Therefore $$S_r = g^{\frac{7}{2}} = g^{3\cdot 5}$$

TABLE I

Thus, with	$g =$	3	3·5	4
We have	$r =$	1·732	1·871	2
And	$S_r = \frac{N_g}{N_l} =$	46·7	80·2	127·9

Case 2.—Two countershaft speeds, $p = 2$, $s = pqm = 16$. Table B gives ($m = 2$, $q = 4$)—

		Speeds.	
	Single gear—		
1	Fast	1 to 4	Step ratio $= r^{1\cdot 5}$
1	Slow	5 to 8	$R_c = \frac{\text{fast}}{\text{slow}}$ countershaft speeds $= r^q = r^4$
	Double gear—		
2	Fast	9 to 12	$R_1 = \frac{D}{C} \times \frac{B}{A} = r^{2q} = r^8$
2	Slow	13 to 16	$R_cR_1 = \quad = r^{3q} = r^{12}$

Thus with $g_1 = g_2 = g$, as before, we have

$$g^2 = r^{2q} = r^8,\ g = r^4, \text{ or } r = g^{\frac{1}{4}}$$

Also $$S_r = \frac{N_g}{N_l} = r^{pqm-1} = r^{15}$$

Therefore $$S_r = g^{\frac{15}{4}}$$

TABLE II

If	$g =$	3	3·5	4
Then	$r =$	1·316	1·368	1·415
And	$S_r =$	61·7	110	182

Case 3.—Three countershaft speeds.

$$p = 3,\ q = 4,\ m = 2,\ s = pqm = 24.$$

Table C gives—

		Speeds.	
	Single gear—		
1	Fast	1 to 4	Step ratio $= r^{1\cdot 5}$
1	Medium	5 to 8	R_{2c} = ratio $\frac{\text{medium}}{\text{slow}}$ countershaft speeds $= r^q = r^4$
1	Slow	9 to 12	R_{3c} = ratio $\frac{\text{fast}}{\text{slow}}$ countershaft speeds $= r^{2q} = r^8$
	Double gear—		
2	Fast	13 to 16	$R_1 = \frac{D}{C} \times \frac{B}{A} = r^{3q} = r^{12}$
2	Medium	17 to 20	$R_{2c}R_1 = \quad = r^{4q} = r^{16}$
2	Slow	21 to 24	$R_{3c}R_1 = \quad = r^{5q} = r^{20}$

Hence $g^2 = r^{3q} = r^{12}, g = r^6$, or $r = g^{\frac{1}{6}}$
And $S_r = r^{23} = g^{\frac{23}{6}} = g^{3.833}$

TABLE III

With	$g =$	3	3·5	4
we get	$r =$	1·200	1·232	1·260
and	$S_r =$	67·45	121·9	207·5

Thus we see that S_r, the ratio of the greatest to the least spindle speed of revolution, increases as the number of countershaft speeds gets greater for a fixed number of cone steps (here = 4), and with one double back gear (*i.e.* $m = 2$); at first rapidly, afterwards more slowly. We find, in fact, that even with a very large number of countershaft speeds (keeping $q = 4$, and $m = 2$) S_r does not increase beyond certain comparatively low limits, viz. 81 for $g = 3$, 150 for $g = 3{\cdot}5$, and 256 for $g = 4$.

This may be easily proved as follows:—

From Table D we have

$$R_1 = g^2 = r^{pq}, \text{ or } g = r^{\frac{pq}{2}}, \text{ or } r = g^{\frac{2}{pq}}$$

Therefore $S_r = r^{pqm-1} = r^{2pq-1} = g^{\frac{4pq-2}{pq}}$

That is, $S_r = g^{4-\frac{2}{pq}}$

which tends to approach the value $S_r = g^4$ as pq continually increases.

But $3^4 = 81, 3{\cdot}5^4 = 150, 4^4 = 256$ *Q.E.D.*

Taking $g = 3\frac{1}{2}$, *i.e.* $g^2 = \frac{D}{C} \times \frac{B}{A} = 12\frac{1}{4}$, as the greatest proper gear ratio for a double gear, we thus see that S_r can never exceed 150; that for three countershaft speeds its value is 122; for two speeds it is 110; and for one speed it is 80.

Two Double Gears

Take now the case of two double back gears; *i.e.* $m = 3$.

Case 1.—One countershaft speed, $p = 1, m = 3, q = 4, s = 12$.

Then Table A reads:—

m	Name of Gear.	Speeds Nos.	Gear Ratios.
1	Single	1 to 4	Step ratio $= r^{\frac{q-1}{2}} = r^{1.5}$
2	First double	5 to 8	$R_1 = \frac{D}{C} \times \frac{F}{E} = r^q = r^4 = g_3 g_2$
3	Second double . . .	9 to 12	$R_2 = \frac{D}{C} \times \frac{B}{A} = r^{2q} = r^8 = g_3 g_1$

Let $\frac{B}{A} = g_1$; $\frac{F}{E} = g_2$; $\frac{D}{C} = g_3$

If $g_2 = 1$
then $g_3 = r^q$
And since $g_3 g_1 = r^{2q} \quad \therefore g_1 = r^q = g^3$
Thus $r = g_1^{\frac{1}{q}} = g_1^{\frac{1}{4}}$
Also $S_r = r^{pqm-1} = r^{11}$
Therefore $S_r = g_1^{\frac{11}{4}}$

TABLE IV

Thus, with	$g_1 =$	3	3·5	4
we get	$r =$	1·316	1·328	1·415
and	$S_r =$	20·51	31·20	45·2

CASE 2.—Two countershaft speeds.

$$p = 2,\ m = 3,\ q = 4,\ s = 24.$$

Table B becomes—

1	Single— Fast	1 to 4	Step ratio $= r^{1\cdot5}$
1	Slow	5 to 8	$R_c = \frac{\text{fast}}{\text{slow}}$ countershaft speeds $= r^q = r^4$
2	First double— Fast	9 to 12	$R_1 = \frac{D}{C} \times \frac{F}{E} = r^{2q} = r^8$
2	Slow	13 to 16	$R_cR_1 = (R_cg_3 \times g_2) = r^{3q} = r^{12}$
3	Second double— Fast	17 to 20	$R_2 = \frac{D}{C} \times \frac{B}{A} = r^{4q} = r^{16}$
3	Slow	21 to 24	$R_cR_2 = (R_cg_3 \times g_1) = r^{5q} = r^{20}$

If $g_2 = 1,\ g_3 = r^{2q}$
and since $g_3 g_1 = r^{4q}$
therefore $g_1 = r^{2q} (= g_3)$
That is $r = g_1^{\frac{1}{2q}} = g_1^{\frac{1}{8}}$
Also $S_r = r^{pqm-1} = r^{23}$
and $S_r = g_1^{\frac{23}{8}}$

TABLE V

Thus, with	$g_1 =$	3	3·5	4
	$r =$	1·148	1·169	1·19
	$S_r =$	23·55	36·73	53·7

Comparing these results for Figs. 38 and 39, we find that Fig. 39 gives values of S_r (*i.e.* ratios of spindle speeds, maximum and minimum) only from ½ to ⅓ as great as those of Fig. 38. [Compare S_r in Tables I. with IV., and Table II. with Table V.] Fig. 39 gives, however, a larger number of speeds than Fig. 26 within these smaller ranges.

[*e.g.* Fig. 38 $s =$ 8 and 16.
Fig. 39 $s =$ 12 and 24.]

We therefore draw the conclusion that the design of Fig. 38 *is more suitable for a general purpose lathe, in which the range of spindle speeds is great, and the number of speed changes small; whilst that of Fig.* 39 *is applicable to work which requires no great range of spindle revolutions, but a finer gradation in the changes of speed, or, in other words, a larger number of them.*

MORE THAN TWO DOUBLE GEARS

Let us now consider the case of more than two double gears.

Let there be any number $(m - 1)$ of double gears; any number (p) of countershaft speeds; and, of course, any number (q) of cone steps. Then the highest gear ratio is (Table D)—

$$\mathrm{R}_{m-1} = r^{p(m-1)q} = g_m \times g_1 = g_1^2, \text{ if } g_m = g_1$$

Therefore $$g_1^2 = r^{p(m-1)q}$$

or $$r = g^{\frac{2}{p(m-1)q}}$$

As before, $$\mathrm{S}_r = r^{pqm-1}$$

Therefore $$\mathrm{S}_r = g^{\frac{2(pqm-1)}{p(m-1)q}} = g^{\frac{2\left(m-\frac{1}{pq}\right)}{m-1}}$$

To find how, in this general case, the value of the spindle-speed ratio changes with p and with m increasing, we may plot curves of the expression—

$$\frac{2\left(m - \frac{1}{pq}\right)}{m - 1}, \text{ or of } \mathrm{S}_r \text{ itself for the more usual values of } g$$

The values of S_r have therefore been calculated for double-geared lathes with one to four back gears, for one, two, three, and four countershaft speeds, and four cone steps. (The same results would be obtained for two cone steps if there were two, four, six, and eight countershaft speeds; so long as the product pq remains the same.)

These values are shown in Table LI., and have been plotted in Figs. 78 and 79.

By following along the *lines* of the table, or the curves of Fig. 78, we can observe how the ratio of greatest to least spindle speed (for double-geared lathes) varies for a given number of double back gears as the number of countershaft speeds is increased.

We see that this ratio increases with increasing numbers of countershaft speeds; at first rapidly, afterwards more slowly [and if pq could be made very great, S_r would approach certain values, viz. 256 for one double gear, 64 for two, and 40·3 for three, without ever reaching them].

By following down the *columns* of the table, on the other hand, or considering the curves of Fig. 79, it is apparent that the spindle-speeds ratio *diminishes* as the number of double gears increases, at first very rapidly, afterwards more slowly [and with m very great, S_r would approach the value g^2 without ever reaching it].

Referring, for example, to Fig. 78; let it be supposed that $\mathrm{S}_r = 30$ (*i.e.* that the ratio of highest number of spindle revolutions is thirty times the

TABLE LI

Values for S_r. Double-geared Lathes.

m	$m-1$	$pq=2$	$pq=4$	$pq=8.$	$pq=12.$	$pq=16.$	$pq=\infty.$
2	1	$3^{3} = 27{\cdot}16$	$3^{3\cdot5} = 46{\cdot}72$	$3^{3\cdot75} = 62$	$3^{3\cdot83} = 68$	$3^{3\cdot875} = 70{\cdot}8$	$3^{4} = 81$
2	1	$3{\cdot}5^{3} = 42{\cdot}8$	$3{\cdot}5^{3\cdot5} = 79{\cdot}43$	$3{\cdot}5^{3\cdot75} = 109{\cdot}6$	$3{\cdot}5^{3\cdot83} = 121$	$3{\cdot}5^{3\cdot875} = 129$	$3{\cdot}5^{4} = 150$
2	1	$4^{3} = 64$	$4^{3\cdot5} = 126$	$4^{3\cdot75} = 182$	$4^{3\cdot83} = 208$	$4^{3\cdot875} = 215$	$4^{4} = 256$
3	2	$3^{2\cdot5} = 15{\cdot}6$	$3^{2\cdot75} = 20{\cdot}75$	$3^{2\cdot875} = 23{\cdot}5$	$3^{2\cdot917} = 24{\cdot}7$	$3^{2\cdot9375} = 25{\cdot}7$	$3^{3} = 27$
3	2	$3{\cdot}5^{2\cdot5} = 22{\cdot}9$	$3{\cdot}5^{2\cdot75} = 31{\cdot}3$	$3{\cdot}5^{2\cdot875} = 36{\cdot}6$	$3{\cdot}5^{2\cdot917} = 38{\cdot}46$	$3{\cdot}5^{2\cdot9375} = 39{\cdot}36$	$3{\cdot}5^{3} = 43$
3	2	$4^{2\cdot5} = 32$	$4^{2\cdot75} = 45{\cdot}4$	$4^{2\cdot875} = 53{\cdot}7$	$4^{2\cdot917} = 57$	$4^{2\cdot9375} = 58{\cdot}9$	$4^{3} = 64$
4	3	$3^{2\cdot3} = 13$	$3^{2\cdot5} = 15{\cdot}6$	$3^{2\cdot58} = 17{\cdot}02$	$3^{2\cdot61} = 17{\cdot}6$	$3^{2\cdot625} = 17{\cdot}86$	$3^{2\cdot66} = 18{\cdot}62$
4	3	$3{\cdot}5^{2\cdot3} = 18{\cdot}6$	$3{\cdot}5^{2\cdot5} = 22{\cdot}9$	$3{\cdot}5^{2\cdot58} = 25{\cdot}12$	$3{\cdot}5^{2\cdot61} = 26$	$3{\cdot}5^{2\cdot625} = 26{\cdot}6$	$3{\cdot}5^{2\cdot66} = 28$
4	3	$4^{2\cdot3} = 25{\cdot}3$	$4^{2\cdot5} = 32$	$4^{2\cdot58} = 35{\cdot}9$	$4^{2\cdot61} = 37{\cdot}3$	$4^{2\cdot625} = 38$	$4^{2\cdot66} = 40{\cdot}27$
5	4	$3^{2\cdot25} = 11{\cdot}9$	$3^{2\cdot375} = 13{\cdot}58$	$3^{2\cdot4375} = 14{\cdot}52$	$3^{2\cdot4583} = 14{\cdot}83$	$3^{2\cdot46875} = 15{\cdot}08$	$3^{2\cdot5} = 15{\cdot}6$
5	4	$3{\cdot}5^{2\cdot25} = 16{\cdot}8$	$3{\cdot}5^{2\cdot375} = 19{\cdot}63$	$3{\cdot}5^{2\cdot4375} = 20{\cdot}8$	$3{\cdot}5^{2\cdot4583} = 21{\cdot}68$	$3{\cdot}5^{2\cdot46875} = 22$	$3{\cdot}5^{2\cdot5} = 22{\cdot}9$
5	4	$4^{2\cdot25} = 22{\cdot}6$	$4^{2\cdot375} = 26{\cdot}92$	$4^{2\cdot4375} = 29{\cdot}3$	$4^{2\cdot4583} = 30{\cdot}06$	$4^{2\cdot46875} = 30{\cdot}84$	$4^{2\cdot5} = 31{\cdot}6$

lowest); then, by going across the horizontal line marked 30, we see that the following five different designs of headstock will fulfil the requirement.

Variation of Sr with number of countershaft speeds and cone steps for lathes with one, two, three or four double gears.

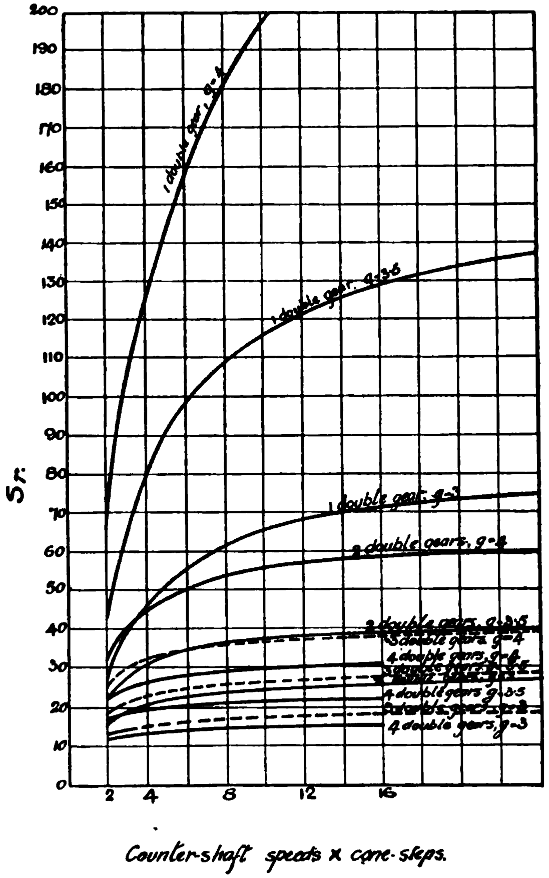


FIG. 78.

Alternative Designs to give $\frac{N_g}{N_l} = 30$.

Number of double gears [$b = (m - 1)$].	Total ratio of said gears (g^n).	Number of counter-shaft speeds (p).	Number of cone steps (q).	Number of spindle speeds ($s = pqm$).
1	(about) 10 to 1	1	2	4
2	,, 15 to 1	1	2	6
2	,, 12 to 1	1	4	12
3	,, 14 to 1	1	4	16
4	,, 17 to 1	2	5	50

There is thus presented a choice of designs, from which the most suitable may be chosen. Similarly for any other spindle-speed range.

LATHES WITH TREBLE-COMPOUND-GEARED HEADSTOCKS

In considering this case we have to remember that of the three pairs of wheels which run in sequence when the highest gear is being used, one pair is usually made available as the first pair of wheels in a double gear drive. Fig. 45, etc., of Chapter XIX., shows such an arrangement. In ascertaining the ratio of spindle speeds which can be obtained by treble gearing, we cannot therefore assume for (g) the greatest constructively convenient or proper value of a pair of gears (spur and pinion, *i.e.*) and make the treble gear equal to the cube of this ratio. The first pair will have a ratio which has reference possibly more to its connection with the lower than with the higher compound gear.

Variation of Sr with number of back gears in double gear lathes with one, two, or three countershaft speeds and two or four cone steps.

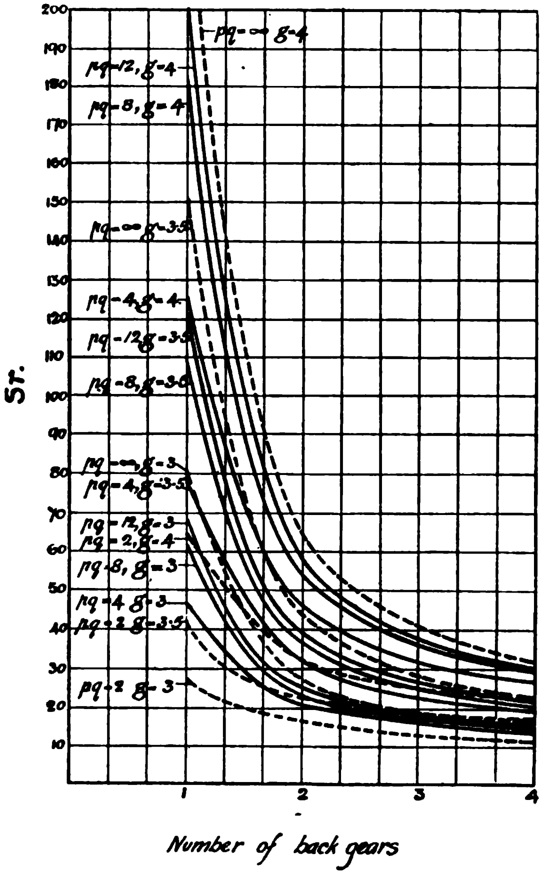


FIG. 79.

We shall first consider a headstock in which the cone is on the spindle, as in Chapter XIX., and in particular Fig. 45. In such headstocks we have usually at least three modes of connection between cone and spindle, viz. direct, double gear, treble gear (*i.e.* $m = 3$). [When the cone is off the spindle, on the other hand, we may have, as in Fig. 50, $m = 2$ only; *i.e.* one single gear, and one treble gear connection of cone to spindle.]

If, then, in Fig. 45, we take $p = 1$, $q = 4$, and $m = 3$, then in Table A we have—

m	Name of gear.	Speeds.	Gear ratio.
1	Direct	1 to 4	Step ratio $r^{1\cdot5}$
2	Double	5 to 8	$R_1 = r^q = \frac{D}{C} \times \frac{B}{A} = g_2 \times g_o$
3	Treble	9 to 12	$R_2 = r^{2q} = \frac{G}{F} \times \frac{E}{C} \times \frac{B}{A} = g_4 \times g_3 \times g_1$

If, now, we took $g_1 = g_3 = g_4$

then $g_1^3 = r^{2q}$

or $r = g_1^{\frac{3}{2q}}$

That is, in this case $r = g_1^{\frac{3}{8}}$

but $S_r = r^{pqm-1} = r^4$

Therefore $S_r = g_1^{\frac{33}{8}} = g_1^{4\cdot125}$

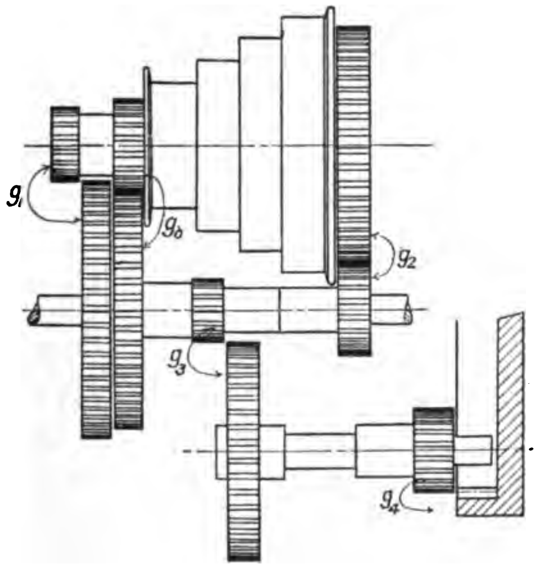


FIG. 80.

Now, if the double gear were, as here supposed, entirely independent of the treble gear, we should have an arrangement such as that shown in the annexed figure (80).

Then $R_1 = g_2 \times g_o = r^q$

but $r = g_1^{\frac{3}{2q}}$

Therefore $g_2 \times g_o = g_1^{\frac{3}{2}}$

And if g_2 were equal to g_o, we should have

$$g_o = g_1^{\frac{3}{4}}$$

TABLE *a*.

Thus with $g_4 = g_3 = g_1$ . .	3	$3\frac{1}{2}$	4	5	6
We should have $g_2 = g_o$ . .	2·28	2·56	2·831	3·344	3·835
r	1·51	1·60	1·683	1·828	1·958
And S_r	93	175	305	766	1667

As matters are arranged in Fig. 45, however, we should find that g_1 would be equal to g_2, and g_3 equal to g_4. Then, again, taking $p = 1$, we have $g_1^2 = r^q$, or $g_1 = r^{\frac{1}{2}q}$. Again, $g_1 \times g_3 \times g_4 = r^{2q}$, or $r^{\frac{1}{2}q} \times g_4^2 = r^{2q}$; *i.e.* $g_4 = r^{\frac{3}{4}q}$. Also $g_4 = g_1 \times r^{\frac{1}{4}q}$. We have, therefore—

$$r = g_4^{\frac{4}{3q}}$$

And since $$S_r = r^{pqm-1}$$

we finally obtain $$S_r = g^{\frac{4(pqm-1)}{3q}} = g_4^{\frac{11}{3}} = g_4^{3{\cdot}666}$$

TABLE *b*.

Thus with $g_4 = g_3$	3	3·5	4	5	6
We should have $g_1 = g_2$.	2·078	2·305	2·52	2·93	3·30
r	1·442	1·519	1·59	1·71	1·82
And S_r	56	99	161	364	716

The advantages of the design of Fig. 45 over the hypothetical arrangement of Fig. 80 are now clearly apparent. It is true that a given spindle speed ratio (*e.g.* 176) can be obtained in the manner shown in Fig. 80, with a smaller value of the greatest velocity ratio of any one pair of wheels than with the design of Fig. 45 (3·5 to 1 for Fig. 80, as against 4·1 to 1 for Fig. 45); but this advantage is but small as against the first cost of the increased number of wheels (10 as against 7) required by Fig. 80. Large ratios, such as 5 or 6 to 1, may be used with impunity when the spur wheel is on the face-plate, and more especially if the gear is internal. The design of Fig. 80, on the other hand, has the advantages that it enables the pitches and proportions of the gears to be better adjusted to their respective requirements, and that the gears will be much more durable, *i.e.* retain their shape, and therefore run smoothly for a longer time than those of Fig. 45. In the latter, the first pair of gears have to be in use at all times, when either the double or the treble gear is in action. Whether these advantages are worth the extra first cost and the extra complication of the gear-changing mechanism, is a problem which can only be answered by practical experience.

With regard to the effect which an increase of the number of treble gears, on the one hand, or an increase in the number of countershaft speeds (or of cone steps) on the other, has upon the ratio of greatest to least spindle speed of revolution, it is to be expected that the same results will be produced as were found above for double gears.

On the assumption that the three pairs of wheels which are in gear when the highest treble gear is in use have all the same velocity ratio (*i.e.* $g_1 \times g_3 \times g_4 = g_1^3$), we find (using Table D) that this highest gear ratio—

$$R_{m \cdot 1} = r^{p(m-1)q} = g_1 \times g_3 \times g_4 = g_1^3$$

Therefore $$g_1 = r^{\frac{p(m-1)q}{3}}$$

or $$r = g_1^{\frac{3}{p(m-1)q}}$$

But $$S_r = r^{pqm-1}$$

Therefore $$S_r = g^{\frac{3(pqm-1)}{pq(m-1)}} = g^{\frac{3\left(m-\frac{1}{pq}\right)}{m-1}}$$

Variation of S_r with number of back gears in treble geared lathes with one, two, or more countershaft speeds and two or four cone steps.

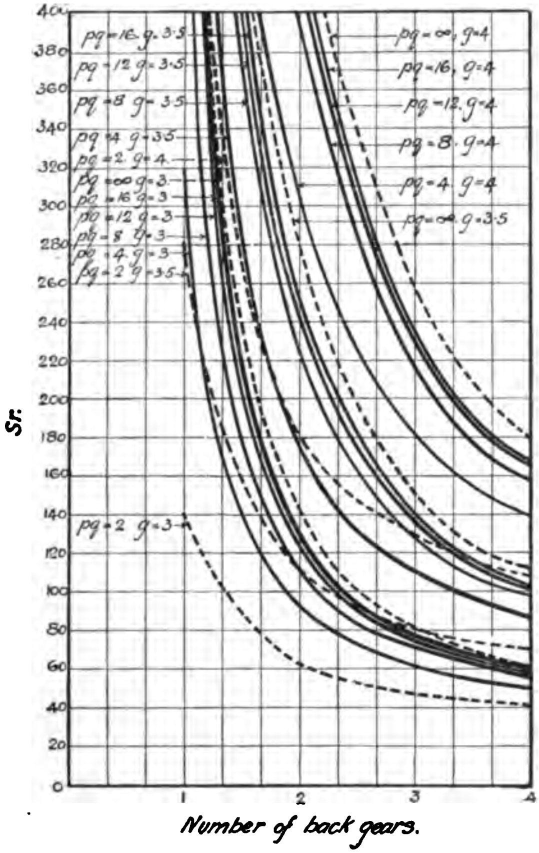


FIG. 81.

The values of S_r obtained from this expression have been calculated, and are tabulated in Table LII., and plotted on Figs. 81 and 82. The same use may be made of these curves, or of the lines and columns of the table, as has already been exemplified for double gears; and it is clear that the effects upon S_r of increasing the number of spindle speeds in the two different ways above specified are of a precisely similar character to those obtained for double gears.

It may be objected that these tables and curves are of but little practical value, by reason of the fact that the first pair of wheels almost always form a component part of a previous or lower treble or double gear. It is easy, however, to allow for this in any given case.

Thus, in the two examples worked out above of Fig. 80 and Fig. 45, we find that for Fig. 80—

$$S_r = g_1^{\frac{3(pqm-1)}{2q}}$$

whilst for Fig. 45 $\quad S_r = g_4^{\frac{4(pqm-1)}{2q}}$

Variation of S_r with number of countershaft speeds and cone steps for lathes with one, two, three, or four treble gears.

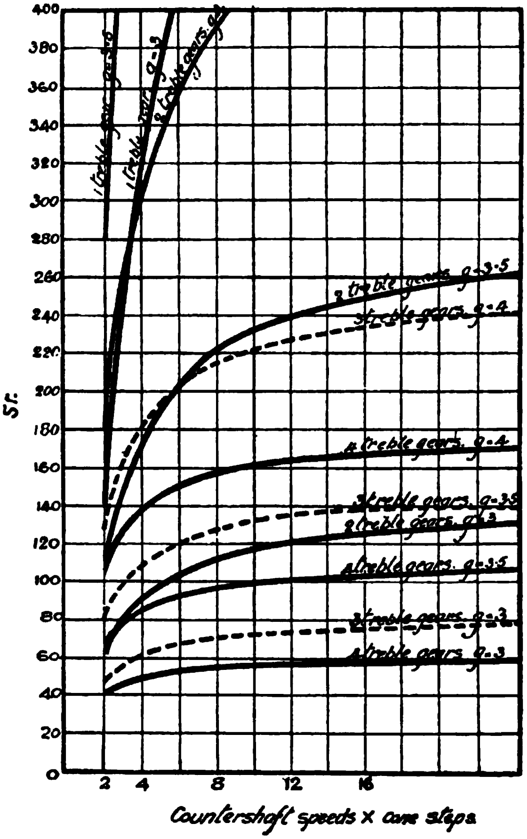


FIG. 82.

If, then, we have found from Table LII. (or Figs. 81 and 82) the most suitable value of g_1 to give any required spindle-speed ratio S_r, we can at once obtain g_4, the higher wheel-pair velocity ratio which is required at the face-plate when the pair of wheels next the cone are used also for the double gear by simply equating—

$$g_1^{\frac{3(pqm-1)}{2q}} \text{ and } g_4^{\frac{4(pqm-1)}{2q}}$$

We then immediately obtain in this instance—

$$g_4 = g_1^{\frac{3}{4}}$$

It is clearly impossible, in the space here available, to pursue, and it is, indeed, doubtful whether any good purpose of practical usefulness would be served by the pursuing of these investigations into more complex arrangements of headstocks.

Enough has been done, at all events, to establish one or two general principles, and to point the way by which any one must travel if he wishes to compare the kinematic advantages of different types of drives; so far at least as changes of the number of countershaft speeds, of cone steps, or of compound gears are capable of influencing them.

It must also be borne in mind that in this investigation we have devoted our attention only to the effect which different designs have upon the ratio of greatest to least spindle revolutions, and upon the number of different spindle speeds which are obtainable with them. Now, this spindle-speed ratio is never required to attain very high values. A reference to Tables XXVI., XXXVII., and XLV., *ante*, will show what these values are likely to be for different types of lathe and kinds of tool steel. The values of S_r are seen to be in general small enough to be of fairly easy attainment in many different ways; and the important consideration arises of the effect upon the belt width, and upon the gear pitch and width, of the various arrangements of drive which we have been considering. This may turn out to be of sufficient importance to decide the question of the design on its merits alone, leaving out of account, as a subordinate matter, the securing of a high spindle-speed ratio. Upon inquiry we find the following state of affairs:—

The general result obtained in the present investigation is that a large number of compound gears is inferior to a large number of countershaft or cone speeds in producing high-spindle speed ratios. We find, however, upon reference to our Chapters X. to XVIII., that precisely the contrary is the case with regard to a good belt drive. *A large number of back gears gives a small belt, whilst a large number of cone steps makes but little difference to the belt!* Thus, in Table XXXIX., we find that for an 18-in. carbon steel lathe, by increasing the number of cone steps from three to four, the belt width is changed only from 9·1 in. to 9·4 in. with three back gears, and from 7·6 in. to 7·5 in. with four back gears; whereas, keeping the number of cone steps the same (say four), and increasing the number of back gears from three to four, the belt width is brought down from 9·4 in. to 7·5 in., *i.e.* by nearly 2 in.

Thus it appears that great care must be exercised when drawing conclusions regarding different types of headstock. Every side of the question must receive its due and proper amount of attention; and the design is of the nature of a compromise which best fulfils the various conflicting conditions of the problem. The effect of different headstock designs upon the economy of machining in the lathe is discussed in Chaps. XL. to XLIII. (below).

ALL-GEAR HEADSTOCKS

In Table L., above, we have given the values of the ratios of the various compound gears in terms of r, p, q, and m for headstocks, with cone pulleys either on or off the spindle. Although this table appears sufficiently elaborate, being carried out so as to include the general case—*i.e.* where

TABLE LII

Values for S_r.

m	$m-1$	$pq = 2$	$pq = 4$	$pq = 8$	$pq = 12$	$pq = 16$	$pq = \infty$
2	1	$3^{4·5} = 140·3$	$3^{5·25} = 319·9$	$3^{5·625} = 482$	$3^{5·75} = 552$	$3^{5·8125} = 589$	$3^{6} = 724·4$
2	1	$3·5^{4·5} = 281$	$3·5^{5·25} = 716$	$3·5^{5·625} = 1148$	$3·5^{5·75} = 1349$	$3·5^{5·8125} = 1479$	$3·5^{6} = 1830$
2	1	$4^{4·5} = 512$	$4^{5·25} = 144·5$	$4^{5·625} = 2438$	$4^{5·75} = 2884$	$4^{5·8125} = 3126$	$4^{6} = 4099$
3	2	$3^{3·75} = 61·4$	$3^{4·125} = 92·5$	$3^{4·3125} = 114$	$3^{4·375} = 121·6$	$3^{4·406} = 126$	$3^{4·5} = 140$
3	2	$3·5^{3·75} = 109·6$	$3·5^{4·125} = 175$	$3·5^{4·3125} = 222$	$3·5^{4·375} = 240$	$3·5^{4·406} = 248$	$3·5^{4·5} = 281$
3	2	$4^{3·75} = 181$	$4^{4·125} = 308·4$	$4^{4·3125} = 394$	$4^{4·375} = 429·5$	$4^{4·406} = 447$	$4^{4·5} = 512$
4	3	$3^{3·5} = 46·8$	$3^{3·75} = 61·4$	$3^{3·875} = 70·3$	$3^{3·916} = 73·6$	$3^{3·9375} = 75·2$	$3^{4} = 80·5$
4	3	$3·5^{3·5} = 80·2$	$3·5^{3·75} = 109·6$	$3·5^{3·875} = 128$	$3·5^{3·916} = 185$	$3·5^{3·9375} = 138$	$3·5^{4} = 149·6$
4	3	$4^{3·5} = 128$	$4^{3·75} = 181$	$4^{3·875} = 215$	$4^{3·916} = 227$	$4^{3·9375} = 234$	$4^{4} = 256$
5	4	$3^{3·375} = 40·7$	$3^{3·562} = 49·9$	$3^{3·656} = 55·4$	$3^{3·687} = 57$	$3^{3·703} = 58·2$	$3^{3·75} = 61·4$
5	4	$3·5^{3·375} = 68·4$	$3·5^{3·562} = 86·5$	$3·5^{3·656} = 97$	$3·5^{3·687} = 101·4$	$3·5^{3·703} = 103·5$	$3·5^{3·75} = 109·6$
5	4	$4^{3·375} = 107·6$	$4^{3·562} = 139$	$4^{3·656} = 158·5$	$4^{3·687} = 165$	$4^{3·703} = 167$	$4^{3·75} = 181$

there are any number (p) of countershaft speeds, any number ($m - 1$) of back gears, and, of course, any number (q) of cone steps—*vide* Table D—it is really not a complete statement of the whole question.

It has been either tacitly assumed or explicitly stated throughout our previous investigations that, in running through the speed changes from the highest to the lowest, the only method allowed or adopted was, that the belt was to be shifted successively from the smallest to the largest step of the cone before making any change in speed of the countershaft. This was described in Chapter VIII. as method A. According to the other method—B of that chapter—the next lower speed is obtained by leaving the cone belt unaltered and shifting the belts on the countershaft pulleys so as to drive that shaft at a slower rate of speed. We there characterized method B as bad practice, on account of the great difference of cone-step diameters which results. This makes it difficult to shift the belt, gives a bad drive on the smallest cone step, and, most important of all, entails an excessively wide belt in order to transmit the required power. We therefore considered it unnecessary to devote our attention to designs involving the method, named B, of obtaining speed changes, but adhered throughout to a discussion of method A alone.

In the case of all-gear heads, however, the alternative method referred to is not to be so lightly dismissed. The three objections mentioned above no longer possess the same validity. There are now neither cone steps nor a belt to shift upon them, and, on the other hand, certain important advantages may be secured by adopting method B.

These methods, redefined to suit the altered circumstances for all-gear heads, may be stated thus:—

(A) On the assumption that there is more than one speed of the countershaft from which the single belt pulley of the all-gear headstock receives its power, a series of successively diminishing speeds may be obtained by changing the gears in the head from the lowest to the highest, whilst all the time keeping the countershaft running at a constant speed. When a still higher speed is required, the lowest gear in the head is reverted to, and the next higher speed of the countershaft is put into operation, the same series of changes of gears in the head being then again repeated, and so on through all the possible changes.

(B) Still on the assumption of more than one countershaft speed, the successive speed variations may be obtained by making all the changes of speed available in the countershaft before resorting to any alteration of gears in the headstock.

It will be found upon investigation that for the purposes of the discussion of these two methods with all-gear headstocks, the tables we gave above are either inapplicable or insufficient. It is, therefore, necessary to introduce a further set here in which the various gear ratios are tabulated as to their values in terms of the geometric ratio, of the number of countershaft—or motor—speeds, and of the various numbers of single, double, treble, or "*n*-tuple" gears. One set will have reference to method A, the other to method B.

In the manner shown by Table XXVII. *et seq.*, in Chapter VIII., we can easily obtain the various values just mentioned. The tables thus compiled, similar to those on p. 42, are somewhat involved, and quite unsuitable for reproduction here. We therefore content ourselves with giving only the results which are required in practice:—

Let there be any number p of countershaft or motor speeds.

Let there be any number a of gears in the first set (single, *e.g.*).

Let there be any number b of gears in the second set (double, *e.g.*).

Let there be any number c of gears in the third set (treble, *e.g.*).

Let there be any number l of gears in the nth or last set (n-tuple, *e.g.*).

Then the following notation and results are afforded:—

TABLE LIII

METHOD A. *All headstock gears used in succession before any change is made in the countershaft speed.*	METHOD B. *All countershaft speed changes made in succession before any change is made in the headstock gears.*
First gear of the 1st set, I. R1 $= g$ Second ,, ,, I. R2 $= gr$ Third ,, ,, I. R3 $= gr^2$ And so on ath gear of the 1st set, I. Ra $= gr^{(a-1)}$	First gear of the 1st set, I. R1 $= g$ Second ,, ,, I. R2 $= gr^p$ Third ,, ,, I. R3 $= gr^{2p}$ And so on ath gear of the 1st set, I. Ra $= gr^{(a-1)p}$
First gear, 2nd set, II. R1 $= gr^{pa}$ Second ,, ,, II. R2 $= gr^{pa+1}$ And so on ath gear, 2nd set, II. Ra $= gr^{(p+1)a-1}$	First gear of the 2nd set, II. R1 $= gr^{ap}$ Second ,, ,, II. R2 $= gr^{(a+1)p}$ And so on bth gear of the 2nd set, II. Rb $= gr^{(a+b-1)p}$
First gear, 3rd set, III. R1 $= gr^{2pa}$ Second ,, ,, III. R2 $= gr^{2pa+1}$ And so on ath gear, 3rd set, III. Ra $= gr^{(2p+1)a-1}$	First gear of the 3rd set, III. R1 $= gr^{(a+b)p}$ Second ,, ,, III. R2 $= gr^{(a+b+1)p}$ And so on cth gear of the 3rd set, III. Rc $= gr^{(a+b+c-1)p}$
And so on	And so on
First gear, nth set, NR1 $= gr^{(n-1)pa}$ Second ,, ,, NR2 $= gr^{(n-1)pa+1}$ And so on ath gear, nth set, NRa $= gr^{[(n-1)p+1]a-}$ $= gr^{(m-a)p+a-1}$ where $m=ap$ the number of ways the driving pulley can be coupled to the spindle.	First gear of the nth set, NR1 $= r^{(a+b+\cdots+k)p}$ Second ,, ,, NR2 $= r^{(a+b+\cdots+k+1)p}$ And so on lth gear of the nth set, NRl $= r^{(a+b+\cdots+k+l-1)p)}$ $= r^{(m-1)p}$ where $m = (a+b+\cdots+l)$ the number of ways the driving pulley can be connected to the spindle.
Also, $c_1 = r^a c_2 = r^{2a} c_3 = \cdots = r^{(p-1)a} c_p$ c_1 = fastest countershaft speed c_2 = next countershaft speed c_p = slowest countershaft speed	Also, $c_1 = rc_2 = r^2 c_3 = \cdots = r^{p-1} c_p$ where c_1 is the fastest countershaft speed ,, c_2 is the next countershaft speed And so on ,, c_p is the slowest countershaft speed
And $S_r = r^{map-1}$	Finally, $S_r = \frac{N_g}{N_l} = r^{(a+b+\cdots+l)p-1}$

In comparing the two schedules of Table LIII., we notice, amongst other differences between method A and method B, one very conspicuous one. This has regard to the number of gears there may be in the different sets.

With B the number of gears in each set may be different for every set. Thus we may have three single gears, four double gears, two treble gears, and so on; whereas, in the case of A, the number must be the same for all. If we have three single gears there must be three double or treble gears, if any, otherwise the countershaft-speed changes which give correctly increasing speeds for the single-gear changes would be unsuitable for the double- or treble-gear changes.

The result of this is clearly seen upon inspection of the schedule of ratios just shown.

As a first numerical example we may take the design of our Fig. 68—Chapter XXI.—omitting, for simplicity, the pair of wheels $\frac{G}{H}$.

We shall then have three single gears, three treble gears, and, if there are two speeds of the countershaft, there will be twelve speeds in all. Let us assume 207 as the highest and 23 as the lowest number of spindle revolutions; then $S_r = 9$.

The calculation then proceeds as follows:—

(A) Adopting method A, *i.e.* assuming all the possible single-gear changes in the headstock are made before resorting to a change of countershaft speed, the same single-gear changes being run through again in succession with the second countershaft speed, and so on.

Then $p = 2,\ n = 2,\ a = 3,\ s = 12$
$S_r = r^{nap-1} = r^{12-1} = r^{11}$
but $S_r = 9$
therefore $r = 9^{\frac{1}{11}}$
from which we easily obtain $r = 1{\cdot}22$

From Schedule A we write down at once

I. $R1 = g =$ say $\frac{65}{27} = 2{\cdot}41$
I. $R2 = gr = 2{\cdot}41 \times 1{\cdot}22 = 2{\cdot}945 = \frac{69}{23}$
I. $R3 = gr^2 = 2{\cdot}41 \times 1{\cdot}486 = 3{\cdot}59 = \frac{72}{20}$

$$c_2 = \frac{c_1}{r^n} = \frac{500}{1{\cdot}22^3} = \frac{500}{1{\cdot}815} = 275{\cdot}3$$

The treble-gear ratios come next, there being no doubles—

II. $R1 = gr^{pa} = 2{\cdot}41 \times 1{\cdot}22^6 = 2{\cdot}41 \times 3{\cdot}297 = 7{\cdot}94 = \frac{53}{16} \times \frac{46}{46} \times 2{\cdot}41$
II. $R2 = gr^{pa+1} = 2{\cdot}41 \times 1{\cdot}22^7$
II. $R3 = gr^{pa+2} = 2{\cdot}41 \times 1{\cdot}22^8$

The speeds are then as follows:—

Single gears.						Treble gears.					
207	167	137	112	92	76	62	51	42	35	28·3	23·2
500			275			500			275		

(B) Adopting method B, *i.e.* running through the successive countershaft speeds before making any change in the headstock, then shifting to the next headstock gear, repeating the countershaft changes, and so on, we have—

$$p = 2,\ a = 3,\ b = 3,\ s = 12 \text{ (as before)},$$
$$S_r = r^{(a+b)p-1} = r^{6\times 2-1} = r^{11};\ \therefore 9 = r^{11},\ r = 1{\cdot}22$$

Using Schedule B, we at once obtain—

$$\text{I. R1} = g = \text{say } \tfrac{65}{27} \text{ as before} = 2{\cdot}41$$
$$\text{I. R2} = gr^{p} = 2{\cdot}41 \times 1{\cdot}22^{2} = 2{\cdot}41 \times 1{\cdot}486 = 3{\cdot}59 = \tfrac{72}{20}$$
$$\text{I. R3} = gr^{2p} = 2{\cdot}41 \times 1{\cdot}22^{4} = 2{\cdot}41 = 2{\cdot}212 = 5{\cdot}32 = \frac{77}{15\frac{1}{2}}$$
$$c_2 = \frac{c_1}{r} = \frac{500}{1{\cdot}22} = 410$$
$$\text{II. R1} = gr^{ap} = 2{\cdot}41 \times 1{\cdot}22^{6} = 2{\cdot}41 \times 3{\cdot}297 = 7{\cdot}94 = \tfrac{53}{16} \times \tfrac{46}{46} \times 2{\cdot}41$$
$$\text{II. R2} = gr^{(a+1)p} = 2{\cdot}41 \times 1{\cdot}22^{8} = 2{\cdot}41 \times 4{\cdot}893$$
$$\text{II. R3} = gr^{(a+2)p} = 2{\cdot}41 \times 1{\cdot}22^{10} = 2{\cdot}41 \times \text{etc.}$$

And the speeds are as before—

207	167	137	112	92	76	62	51	42	35	28·3	23·2
500	410	500	410	500	410	500	410	500	410	500	410

with a different lower countershaft speed.

As a second example we may take the arrangement shown in Fig. 69 of Chapter XXI., where we have two single gears and eight treble gears. Method B is obligatory in this case, as the two sets have different numbers of gears. Let $p = 2$. In Fig. 69

$$s = mp = (a + b)p = (2 + 8)2 = 20 \text{ speeds}$$

Let $N_g = 207$, $N_t = 23$, $S_r = 9$

Then $$S_r = r^{(a+b)p-1} = r^{19}$$
or $$r = 9^{\frac{1}{19}} = 1{\cdot}123$$
for $$\log 9 = 0{\cdot}9542$$
$$\log r = \frac{0{\cdot}9542}{19} = 0{\cdot}05024 = \log 1{\cdot}123$$

Let $g = 2$

$$\begin{array}{r} \log r = 0{\cdot}05024 \\ 2 \\ \hline 0{\cdot}10048 = \log 1{\cdot}26 \end{array}$$

Then $\quad \text{I. R1} = 2 = \dfrac{\text{B}}{\text{A}}$ (*vide* Schedule B)

$$\text{I. R2} = gr^{p} = 2 \times 1{\cdot}123^{2} = 2 \times 1{\cdot}26 = 2{\cdot}52 = \frac{\text{D}}{\text{C}}$$

Also $\quad \text{II. R1} = gr^{ap} = 2 \times 1{\cdot}26^{2\times 2} = 2 \times 1{\cdot}26^{4}$

$$\begin{array}{r} \log r = 0{\cdot}05024 \\ 4 \\ \hline 0{\cdot}20096 = \log 1{\cdot}588 \end{array}$$

$$= 3{\cdot}176$$

Thus $\quad \dfrac{\text{H}}{\text{G}} \times \dfrac{\text{F}}{\text{E}} \times \dfrac{\text{B}}{\text{A}} = 3{\cdot}176 = \dfrac{\text{H}}{\text{G}} \times 1 \times 2$

or $\quad \dfrac{\text{H}}{\text{G}} = 1{\cdot}588$

Again, $\quad \text{II. R2} = gr^{(a+1)p} = 2 \times 1{\cdot}26^{6}$

$$\begin{array}{r} \log r = 0{\cdot}05024 \\ 6 \\ \hline 0{\cdot}30144 = \log 2{\cdot}002 \end{array}$$

$$= 4{\cdot}004$$

Therefore $\quad \dfrac{\text{H}}{\text{G}} \times \dfrac{\text{F}}{\text{E}} \times \dfrac{\text{D}}{\text{C}} = 4 = 1{\cdot}588 \times 1 \times 2{\cdot}52 = 4$ (as it ought to be)

Again $\quad \text{II. R3} = gr^{(a+2)p} = 2 \times 1{\cdot}26^{8}$

$$\begin{array}{r} \log r = 0{\cdot}05024 \\ 8 \\ \hline 0{\cdot}40192 = \log 2{\cdot}523 \end{array}$$

$$= 5{\cdot}046$$

Therefore $\quad \dfrac{\text{J}}{\text{I}} \times \dfrac{\text{F}}{\text{E}} \times \dfrac{\text{B}}{\text{A}} = 5{\cdot}046 = \dfrac{\text{J}}{\text{I}} \times 1 \times 2$

or $\quad \dfrac{\text{J}}{\text{I}} = 2{\cdot}523\left(= \dfrac{\text{D}}{\text{C}}\right)$

Also $\quad \text{II. R4} = gr^{(a+3)p} = 2 \times 1{\cdot}26^{10}$

$$0{\cdot}5024 = \log 3{\cdot}180$$

$$= 2 \times 3{\cdot}180 = 6{\cdot}36$$

Therefore $\quad \dfrac{\text{J}}{\text{I}} \times \dfrac{\text{F}}{\text{E}} \times \dfrac{\text{D}}{\text{C}} = 6{\cdot}36 = 2{\cdot}523 \times 1 \times 2{\cdot}523 = 6{\cdot}36$ (as it should be)

and so on, $\quad \text{II. R}b = gr^{(a+b-1)p} = 2 \times 1{\cdot}26^{(2+8-1)2} = 2 \times 1{\cdot}26^{18} = 16{\cdot}05$

We thus obtain $\quad \dfrac{\text{B}}{\text{A}} = 2;\ \dfrac{\text{F}}{\text{E}} = 1;\ \dfrac{\text{H}}{\text{G}} = 1{\cdot}588;\ \dfrac{\text{J}}{\text{I}} = 2{\cdot}53 = \dfrac{\text{L}}{\text{K}} = \dfrac{\text{D}}{\text{C}}$

Also $\quad c_2 = \dfrac{c_1}{r} = \dfrac{500}{1{\cdot}26} = 396$

If we attempted to use method A in this design, where there are eight treble and only two single gears, the second countershaft speed, which would be $c_2 = \frac{c_1}{r_a}$, and which ought to fit in as the third change after the two single gears in the head had been run through, would obviously not be correct for a following-on speed after the eight treble gears had been used with the first speed of countershaft.

This seems a mere truism when looked at in this way, and is possibly well known to experts; but to the authors it came as a surprise, having only been discovered to them by their analysis in connection with Table LIII.

CHAPTER XXIII

DESIGN AND PROPORTIONS OF POWER GEARS

Two principal considerations govern the design of wheel teeth. They are strength and durability. The former depends upon the forces the teeth have to transmit, and also, indirectly, upon the speed of rotation, since with high speeds the teeth are more subject to shocks or blows, due to irregularity of motion. The latter depends not so much upon the maximum force to be transmitted as upon the number of hours in the working year the wheel is employed in transmitting power, and the proportion of the average to the maximum power so transmitted. In the rational design of gears we must, therefore, be prepared to consider not only the greatest torque which will act upon the wheel teeth, but also the number of times per week or year the load due to that torque will come upon any given tooth when in work, and the proportion which the time the wheel is idle bears to the time it is working.

With the advent of high-speed steels the conditions governing durability have become more important than ever before. Now that the speeds of cutting are so much increased, the provision of a geometrically correct form of tooth, to begin with, and the maintenance of that form during the life of that tooth, are absolutely essential for the preservation of the gearing from injury by shocks, and for ensuring a smooth finish upon the material.

Formulæ for Strength.—Of the three possible classes of gears, viz. machine-cut, machine-moulded, and those cast from a pattern, we propose to consider only the first two, as no pretence to accuracy can be obtained with the last-named wheels.

When the teeth of wheels are machine cut, their shafts correctly proportioned for rigidity, and the bearings of these shafts of sufficient area and in good alignment, it may be taken for granted that the teeth of a pair of enmeshing wheels have a bearing along the whole width of the tooth, and that the load is uniformly distributed over the whole of that length.

For the engineer the problem, then, becomes the simple one of a cantilever of span h, breadth b, depth d, loaded at the end with a force P. If f is the stress produced at the dangerous section—which may or may not be at the root of the tooth—we have—

$$Ph = \frac{bd^2f}{6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

The load does not always act at the outer edge of the tooth, and very often more than one tooth on each wheel is in gear at once, so that only a certain fraction, say $1/e$th of the bending moment Ph, really acts upon a tooth. In that case we have—

$$\left.\begin{aligned}\frac{Ph}{e} &= \frac{bd^2f}{6} \quad \dots\dots\dots\dots \\ \text{or} \qquad P &= \frac{ed^2bf}{6h} \quad \dots\dots\dots\dots\end{aligned}\right\} \quad (74)$$

For any given material, velocity ratio and form of tooth, the proportion of depth or thickness of tooth at the root and of span or height of tooth to the pitch of the teeth is a constant.

Let $d = ap$, and $h = \beta p$. Then (74) may be written

$$P = \frac{ea^2p^2}{6\beta p}bf = \frac{ea^2}{6\beta}pbf \quad \dots\dots\dots \quad (75)$$

Putting y for $\frac{ea^2}{6\beta}$ we obtain the well-known and much-employed formula of Lewis—

$$P = ypbf \quad \dots\dots\dots\dots \quad (75\text{A})$$

For the values of y and f as worked out by Lewis from a series of drawings of teeth for the involute, cycloidal, and radial flank system, see the tables given below (from "Kent's Pocket-book"). By means of these drawings Lewis determined the position of the dangerous section for each variety, and the ratios to the pitch of the tooth-height and of the tooth-thickness at that section, and hence the values of a and β, that is y, were known. Throughout the table Lewis assumed $e = 1$, *i.e.* only one tooth to be in gear at a time.

TABLE LIV

Table of Lewis' Values of y.

Number of teeth.	Involute (20 deg. obliquity).	Involute (15 deg. obliquity) and cycloidal.	Radial flanks.
12	0·078	0·067	0·052
13	0·083	0·070	0·053
14	0·088	0·072	0·054
15	0·092	0·075	0·055
16	0·094	0·077	0·056
17	0·096	0·080	0·057
18	0·098	0·083	0·058
19	0·100	0·087	0·059
20	0·102	0·090	0·060
21	0·104	0·092	0·061
23	0·106	0·094	0·062
25	0·108	0·097	0·063
27	0·111	0·100	0·064
30	0·114	0·102	0·065
34	0·118	0·104	0·066
38	0·122	0·107	0·067
43	0·126	0·110	0·068
50	0·130	0·112	0·069
60	0·134	0·114	0·070
75	0·138	0·116	0·071
100	0·142	0·118	0·072
150	0·146	0·120	0·073
300	0·150	0·122	0·074
Rack	0·154	0·124	0·075

$$y = \left(0{\cdot}124 - \frac{0{\cdot}684}{T_1}\right)$$

TABLE LV

Lewis' Values of f.

Speed in feet per minute.	100 or less.	200	300	600	900	1200	1800	2400
Cast iron . .	8,000	6,000	4,800	4,000	3,000	2,400	2,000	1,700
Steel . . .	20,000	15,000	12,000	10,000	7,500	6,000	5,000	4,300

The values of e for $h = 0·55p$ are given in the accompanying table, and plotted in Fig. 83. To obtain the values of e for any other proportion of height to pitch of tooth, the following expression, in which r is the velocity ratio of the gear, may be used :—

$$e = \beta\sqrt{\frac{rT_1}{1+r}}$$

TABLE LVI

Values of e.

Gear ratio $r = \left(\frac{T_2}{T_1}\right)$	Number of teeth in pinion (T_1)			
	10	15	20	25
1 to 1	1·23	1·504	1·74	1·945
2 to 1	1·42	1·738	2·01	2·245
4 to 1	1·555	1·905	2·2	2·468
6 to 1	1·61	1·972	2·28	2·545
8 to 1	1·64	2·01	2·32	2·59
10 to 1	1·66	2·03	2·345	2·62

From this table we perceive the advantage of keeping the velocity ratio high, the number of teeth large, and the pitch fine if a large fractional engagement is desired. The velocity ratio is, however, otherwise limited by the costliness of the larger sizes of spur wheels, so that a large velocity ratio really means a small pinion with few teeth, which, as the table shows, gives poor values of e; so that a fine pitch, or a large number of teeth, is really the desideratum to be aimed at.

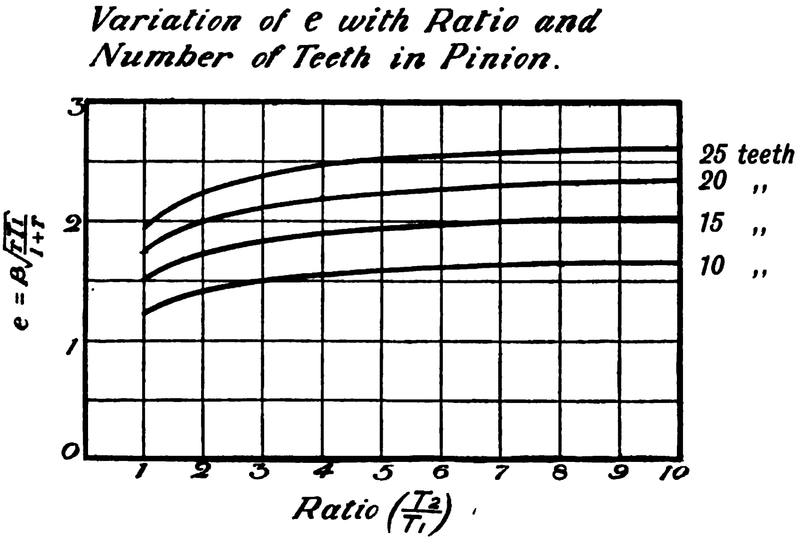


FIG. 83.

The authors' experience with Lewis' formula points to its giving excessively strong teeth at low speeds of revolution. The following cases may be cited as examples of this:—

(1) A face-plate pinion of machine-cut cast iron for a large lathe had 16 teeth of 3-in. pitch, 7½-in. width, and loaded with 42,000 lb., the speed being 18 ft. per minute at the pitch circle. According to Lewis, the permissible load on such a pinion is only 14,000 lb., and the pitch it would have had if his formula had been followed is 9 in. Even allowing a breadth of 12 in., the pitch would have had to be 5·7 in.

(2) A face-plate pinion of mild steel, machine-cut, had 14 teeth of 2¾-in. pitch, with a width of 8¼ in., to carry a load of 85,000 lb. at 28 ft. per minute. The safe load, according to Lewis, would only be 32,700 lb. According to the formula developed further on, we should have for these two cases the following proportions:—

Case I.—To find the pitch, the breadth being determined by formula (79) on p. 157.

$$p = \left(\frac{\mathrm{P}\sqrt{\mathrm{N}_1}}{3000be}\right)^2 = \left(\frac{42{,}000\sqrt{4{\cdot}5}}{3000 \times 7{\cdot}5 \times 2}\right)^2 = 3{\cdot}9 \text{ in.}$$

Or by formula (81)

$$p = \frac{\mathrm{P}\sqrt{\mathrm{N}_1}}{12{,}000e} = \frac{42{,}000 \times \sqrt{4{\cdot}5}}{12{,}000 \times 2} = 3{\cdot}7 \text{ in.}$$

Case II.—

$$p = \left(\frac{\mathrm{P}\sqrt{\mathrm{N}_1}}{9000be}\right)^2 = \left(\frac{85{,}000\sqrt{9}}{9000 \times 8{\cdot}25 \times 2}\right)^2 = 2{\cdot}95 \text{ in.}$$

By formula (81)

$$p = \frac{\mathrm{P}\sqrt{\mathrm{N}_1}}{36{,}000e} = \frac{85{,}000\sqrt{9}}{36{,}000 \times 2} = 3{\cdot}54 \text{ in.}$$

From these and many other examples it appears that Lewis' formula is in error, in that it gives results in excess of the requirements, more especially at the slower speeds.

The following formula has been constructed by the authors for the purpose of designing gears for lathe headstocks. It is based upon an exhaustive comparison of over two hundred examples taken from practice during the last few years. It gives results which are in accordance with the most modern types of headstock design, both British and foreign, but gives widths of tooth which are somewhat less than those which the experimental and theoretical investigations of Stribeck and Lasche would lead to.

We begin with the rational formula (75), viz.—

$$\frac{\mathrm{P}}{epb} = \frac{a^2}{6\beta}f$$

$\frac{a^2}{6\beta}$ [for which Lewis wrote y, and gave the value $y = \left(0{\cdot}124 - \frac{0{\cdot}684}{\mathrm{T}_1}\right)$ for 15 deg. involute and cycloidal wheels up to fifty teeth—beyond which this value scarcely alters, so that the value for fifty teeth may be taken] may be expressed as follows:—

$$y = \frac{a^2}{6\beta} = 0{\cdot}019\sqrt{\mathrm{T}_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

where T_1 is the number of teeth in the pinion of the pair of gears under consideration. Also the value of the allowable safe stress f is found to vary inversely as the square root of the speed of the pitch circle, since the greater this speed the greater the vibration and shock. Thus

$$f = \frac{c}{\sqrt{V}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

Hence
$$\frac{P}{epb} = yf = \frac{0{\cdot}019\sqrt{T_1}c}{\sqrt{V}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

Now $V = T_1 pN_1$, where N_1 is the number of revolutions of the pinion. So that

$$\frac{P}{cpb} = \frac{0{\cdot}019\sqrt{T_1}c}{\sqrt{T_1}\sqrt{pN_1}} = \frac{0{\cdot}019c}{\sqrt{pN_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (78\text{A})$$

or
$$\frac{P\sqrt{N_1}}{be\sqrt{p}} = 0{\cdot}019c = C = 3000 \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

$$p = \left(\frac{P\sqrt{N_1}}{3000be}\right)^2$$

for cast iron machine-cut pinions.

In using the formula (79) to find the value of p for a given P, N, and e for any material, the ratio of b to p must be assumed. The tendency of modern practice is to increase the ratio of width to pitch as the pitch becomes finer, and we have found the rule

$$b = (2\tfrac{1}{2} + p)\sqrt{p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

to be most generally in agreement with the best practice. The values of breadth relatively to pitch so deduced are given in the accompanying table.

$p = \frac{1}{2}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4
$b = 2{\cdot}12$	3·5	4·9	6·85	7·9	9·5	11·2	13

The formula may then be written

$$\frac{P\sqrt{N_1}}{e(2\frac{1}{2} + p)p} = C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

$= 3000$ for cast iron
$= 6000$ for phosphor bronze
$= 9000$ for mild steel

or
$$\frac{P\sqrt{N_1}}{3000e} = (2\tfrac{1}{2} + p)p \text{ for cast iron}$$

$$\frac{P\sqrt{N_1}}{3000e} + \frac{25}{16} = (p + \tfrac{5}{4})^2 \text{ for cast iron}$$

Hence
$$\frac{\sqrt{(P\sqrt{N_1} + 4700)}}{55\sqrt{e}} - 1{\cdot}25 = p \text{ for cast iron} \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

and
$$\frac{\sqrt{(P\sqrt{N_1} + 9400)}}{77{\cdot}5\sqrt{e}} - 1{\cdot}25 = p \text{ for phosphor bronze}$$

and
$$\frac{\sqrt{(P\sqrt{N_1} + 14{,}000)}}{95\sqrt{e}} - 1{\cdot}25 = p \text{ for mild steel}$$

This may also be written in the form—

$$T_1 = \frac{173D\sqrt{e}}{\sqrt{(P\sqrt{N_1} + 4700)} - 69\sqrt{e}} \text{ (for cast iron) . . } (83)$$

from which the number of teeth is known when the load, the revolutions, and the fractional engagement are given, and the diameter has been decided upon.

The formula written in this way leaves somewhat to be desired in regard to simplicity. For any other desired values of b/p the formula (79) may, however, be employed.

Another Method.—The method recommended by Bach, Stribeck, and Lasche, as the result of their investigations, is first to determine the width of the tooth from the consideration of its durability. A tooth of a given pitch can only be made more durable by increasing its width; increasing the pitch diminishes its wearing qualities. This we shall now proceed to discuss more closely.

Formulæ for Durability.—The surfaces of two teeth in contact when transmitting power would, if incompressible, touch along a line having no measurable breadth, and of length equal to the width of the teeth. As, however, the material of the surfaces must yield somewhat, the load on the teeth will act over a narrow surface of finite area, and the pressure produced per unit of that area will be more intense the narrower it is. Experimental data regarding the widths assumed by this contact area, with different shapes of teeth and under different loads, are, so far as the authors know, entirely lacking; as also is any information regarding the action of a lubricant, which, by remaining as a thin film, even under intense momentary pressures, between these contact surfaces, may—under favourable conditions, as to small frequency of contact, ready renewal of the film, and not very dissimilar surface curvature—very greatly reduce the amount of metallic contact, and consequent wear of the teeth. Neglecting, however, any lubrication effect, it is next found that from the point of view of the securing of a durable form of tooth, the amount and speed of the relative movement of the one surface over the other is just as important as the width of that surface.

The amount of abrasion of the tooth depends even more on the amount of sliding which takes place than upon the pressure intensity under which the load is transmitted.

The study of this problem has received much attention of late in Germany, and engineering literature has been enriched by the contributions of Stribeck, Bach, and Lasche upon the subject. Stribeck's work was published in the *Zeitschrift des Vereins Deutscher Ingenieure* for 1894, pp. 168 and 1187. Lasche's appeared in the same journal for 1899, pp. 1417 *et seq.* Bach's results are to be found in his treatise on "Machine Elements."

All we have space for here is a short summary of results. If the wear of a tooth is proportional to the specific pressure (*i.e.* load divided by area of contact surface at any instant), due to the load and to the amount of relative sliding jointly, then some attempt must be made to determine these quantities for teeth of different shape and for the faces and flanks of any one given tooth. The amount by which two cylindrical surfaces will compress each other, which are pressed together whilst their axes are kept parallel, is some function of the difference of their curvatures. The area of the "flattening," when two cylinders convex to each other are pressed together, will obviously be smaller than when a convex presses upon a plane or upon a concave surface. We may therefore expect that the area of the mutual surfaces of contact will be greater, and the specific pressure upon them therefore less, for cycloidal than for involute teeth, for the flanks of the former are usually hollow, and the convex faces of the opposing teeth a better fit upon them than is the case with the latter. Involute teeth, being of convex-outwards shape over their whole working surfaces, are less favourably circumstanced in this respect.

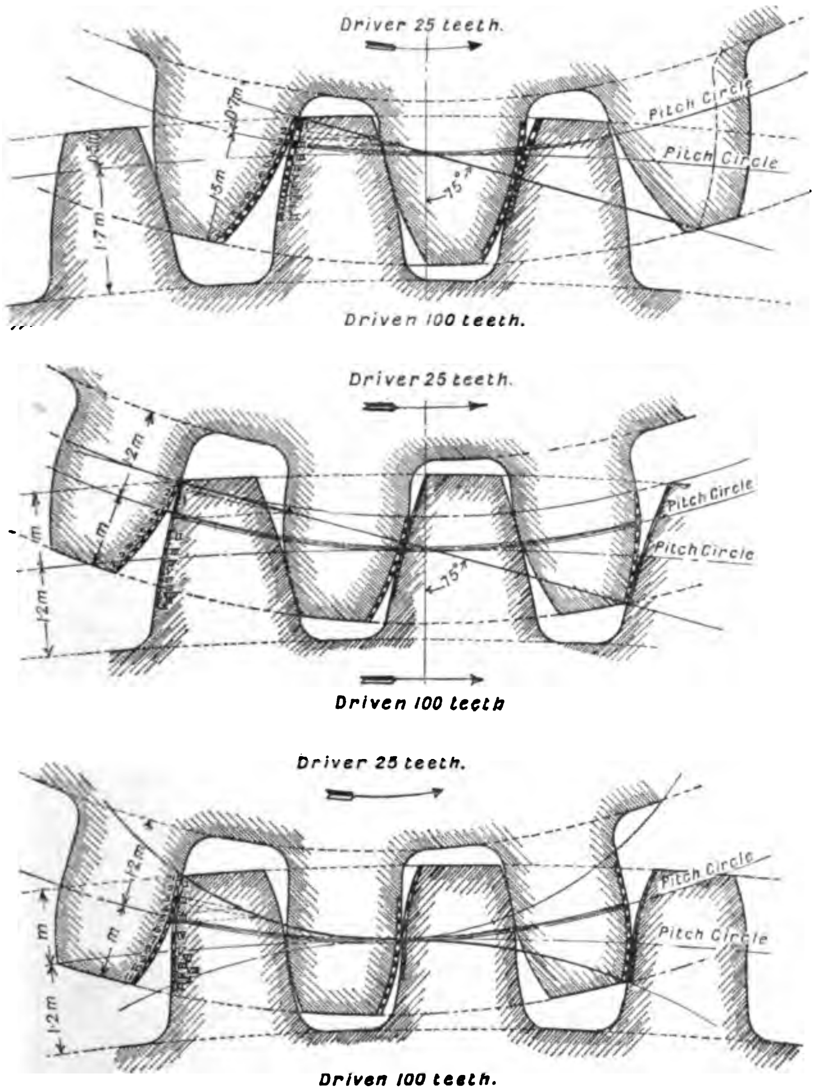
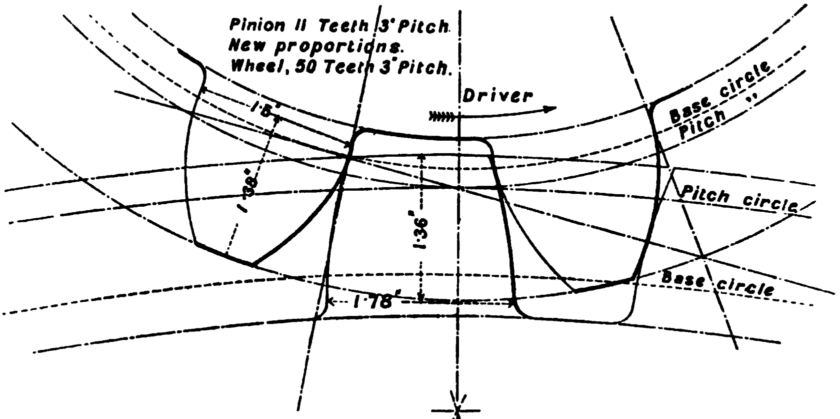


FIG. 84.

The amount of relative sliding of the various parts of the faces and flanks of the two varieties of teeth has been carefully studied by Lasche. For this purpose he divided the working profile of the driving tooth—Fig. 84—into a number of equal portions marked 1, 2, 3, etc. He then determined the lengths of the corresponding portions I., II., III., etc., upon the driven tooth. The ratio of the lengths of the profile portions which work together then gives the amount of the relative sliding.

FIG. 85.



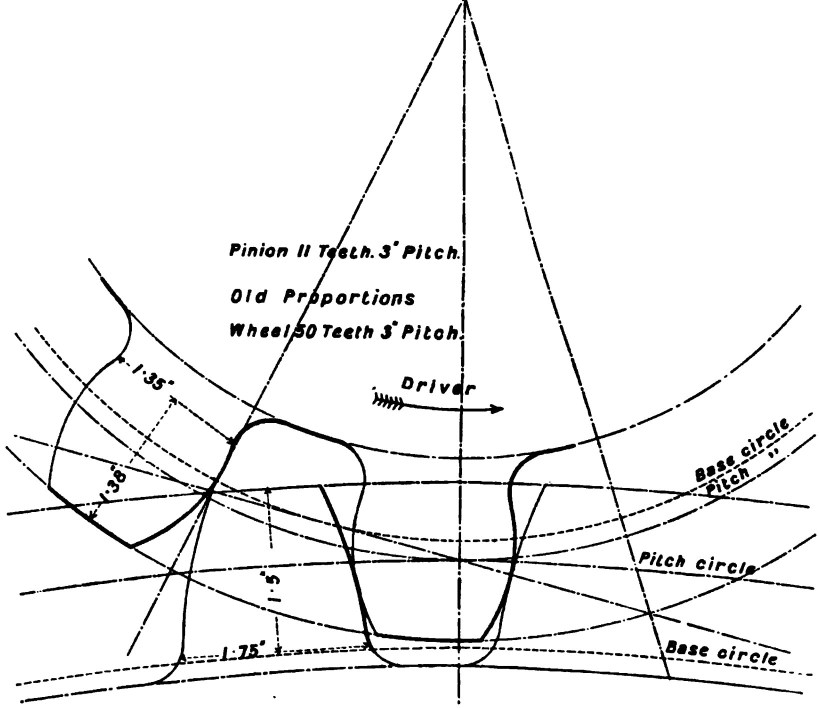


FIG. 86.

In the case of cycloidal teeth the amount of sliding is the same for all portions upon the same side of the pitch circle.

For involute teeth the relative sliding is found—Fig. 84—to be exceptionally great at the point of the driven, or larger, wheel. It diminishes towards the root, at first rapidly, afterwards more slowly.

Lasche consequently finds that for teeth of the ordinary lengths above and below the pitch circle the involute tooth is inferior in wearing quality to the cycloidal. *Notwithstanding this, he recommends,* as the final result of his work, *that involute teeth be always employed in cases where the preservation of tooth form is of importance.* Fig. 84 will at once explain how this conclusion comes to be the correct one. It is there seen that by shortening the face of the driven and the flank of the driving wheel, and *vice versâ,* the obnoxiously large amount of relative sliding of the point of the driven wheel over the flank of the driver during the arc of approach is got rid of, and as a net result *the average amount of relative sliding taken over the whole tooth surface is for involute teeth reduced below that for cycloidal teeth.* The length of the arc of approach is, of course, diminished, but the total arc of contact is maintained the same as usual. A lengthening of the arc of recess is produced by the increased projection of the face of the driving tooth beyond the pitch circle.

Teeth formed in this way are shown dimensioned in Fig. 86, and for comparison, the older proportions as usually employed are given in Fig. 85. This highly efficient and very durable form of tooth has been recently adopted by Messrs. Joseph Adamson and Co., of Hyde, for their electric cranes, and a comparison of the two systems made by them is, by their kind permission, given in the accompanying tables LVII. and LVIII.

TABLE LVII

Table showing Increased Strength due to New Proportions.

Number of teeth.	Percentage stronger compared with old proportions.	
	Driver.	Driven.
12	34·5	0·0
15	31·0	1·0
18	26·0	2·25
21	16·0	3·5
27	12·0	5·0
40	8·0	8·0
80	8·5	20·0
Rack	18·0	31·0

TABLE LVIII

Number of teeth in wheel.	Number of teeth in pinion.	Arc of contact.		Pitch.	Teeth in gear $= \frac{\text{arc of contact}}{\text{pitch}}$.	
		Old proportions.	New proportions.		Old proportions.	New proportions.
		in.	in.	in.		
50	11	4·3	3·44	3	1·43	1·15
50	12	4·37	3·49	3	1·46	1·16
50	14	4·44	3·58	3	1·48	1·19
50	16	4·5	3·67	3	1·50	1·22
50	18	4·55	3·74	3	1·52	1·24
50	20	4·6	3·8	3	1·53	1·27

Old proportions.

Wheel teeth = 0·25 of pitch above pitch circle.
Pinion teeth = 0·25 of pitch above pitch circle.
Clearance = 0·07 of pitch.
Total height of tooth = 0·570 of pitch.

New proportions.

Wheel teeth = 0·100 of pitch above pitch circle.
Pinion teeth = 0·350 of pitch above pitch circle.
Clearance = 0·050 of pitch.
Total height of tooth = 0·500 of pitch.

The advantages claimed by Herr Lasche for this form of tooth are very marked, and should, if well substantiated, lead to their more general adoption. In addition to the diminution of the wear to which they are subject, they are of much stronger form for a given pitch than ordinary involutes, and still more so than ordinary cycloids. They are also more easily machine cut, as the flank portions are more open and accessible to the milling cutter, and the cutter is more easily made and kept. There is also, of course, the advantage possessed by all involute teeth over cycloidal, that alteration of the distance between the centres of the wheels, due to wear of the shaft bearings, does not affect their smooth running.

In case of the adoption of Lasche's form of teeth it must be remembered that the constants given in tables LIV. and LV. above for Lewis's formula are not applicable. They will give results much larger than necessary; the teeth, *i.e.*, would be much too strong if calculated by the ordinary formula. The rational formula (75) is, however, available for the purpose by putting in the actual values of a and β, and employing a suitable value for f, the sizes of the new form of tooth can readily be determined.

For teeth of given form, the amount of wear in a given time, say one working year, will be proportional not only to the above-mentioned factors, load per unit of contact area, and amount of relative sliding; but also, of course, to *the number of contacts made* by a tooth of the smaller wheel *in that given time.*

This must therefore be borne in mind in designing teeth for wear; as a pinion which stands idle half the year need not obviously have so great a breadth, even if it have on occasion to transmit equal power, as one which is constantly employed in conveying work for all the gears indiscriminately.

The formula proposed by Stribeck, and endorsed by Lasche, for determining b, the width of the teeth of a given pinion, *i.e.* pair of wheels, which makes N revolutions per minute and transmits an average load P, is—

$$\frac{\mathrm{PN}}{eb} = \text{constant}$$

This may be seen to follow from the assumptions above mentioned—that the wear is proportional to the specific pressure, the relative sliding and the number of contacts per minute, as follows.

Let e be the number of teeth in contact at any one time;[1] then $\frac{\mathrm{P}}{e}$ is the load taken by any one tooth, on the assumption that that load is shared equally by the e teeth in contact. If Δx is the (unknown) breadth of the above described "flattening," or surface of mutual contact, then the specific pressure is $\frac{\mathrm{P}}{eb\Delta x}\cdot$ The relative sliding is a number obtained by dividing the lengths of the little arcs I., II., III. (Fig. 84) by the lengths of their corresponding arcs 1, 2, 3, on the driving and driven wheels. This number depends only on the form of tooth and its height above and depth below the pitch circle. Let it be denoted by the letter z. Then the wear is proportional to $\frac{\mathrm{P}}{eb\Delta x}$ and to z conjointly when the wheels run at a given speed.

---

[1] Here e is not necessarily integral. Comp. Lasche *loc. cit.*

Lastly, since any pinion tooth has as many contacts per annum with the teeth of its mating spur wheels as it makes revolutions per year, the wear will vary as cN where N is the turns per minute the pinion makes when in gear, and c is the ratio of the hours it works per annum to the total hours in the working year (say 3000).

If, now, the wear is not to exceed a certain amount (θ) inches per annum, we must choose b so that the expression $\frac{zPcN}{eb\Delta x}$ (to which that wear is proportional) shall not be greater than a certain numerical value, fixed by practical experience, or special experiments. In other words—

$$\frac{zPcN}{eb\Delta x} \overset{=}{<} \theta$$

or

$$\frac{PcN}{eb} = \frac{\theta\Delta x}{z} = K^1 = \text{a constant}$$

For any given ratio of diameters of spur to pinion, and for a given tooth form, e is known. For ordinary lathe headstock first-motion pinions, for example, e does not differ much from z, when the height of the tooth is 0·55 of the pitch. Thus we may write for such gears—

$$\frac{PN}{eb} = K = \begin{cases} 40{,}000 \text{ for cast iron, machine cut} \\ 100{,}000 \text{ for steel, machine cut} \end{cases}$$

Procedure in designing Gears in general.—First find P, the load on the teeth; e, the number of teeth in gear at any one time; and N, the number of revolutions per minute the smaller wheel makes; also c the ratio of time working to time idle.

Then the breadth is determined from—

$$b = \frac{PcN}{eK^1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

The pitch must now be determined by using the rational formula (75), which may be written

$$\frac{P}{epb} = \frac{a^2}{6\beta}f = L, \text{ say}$$

The constant L depends upon: a the ratio of thickness to pitch, β the ratio of height to pitch, and f the safe allowable stress. As already mentioned, f depends upon the speed; because the higher the speed the greater the shock upon the tooth—that is, the greater the value of the alternation of the stress, and the greater the rapidity of its repetition. Wöhler has shown that fatigue of the material sets in, or, in other words, the life of the tooth will be shorter the more rapidly the stress alternates, and the larger the difference between its greatest and least values. Bach has put forward the suggestion $f = A - B\sqrt{N}$ for the value of the safe stress where A and B are constants depending on the material and nature of the loading, and N is the number of revolutions per unit of time of the pinion.

In what follows we propose to adopt the form $f = \frac{K}{\sqrt{N}}$ on account of its greater simplicity.

The values of $\frac{a^2K}{6\beta}$, which give the best agreement with recent practice, are given in the following statement. For cast-iron gears, machine-cut—

$$\frac{P}{epb} = \frac{a^2K}{6\beta\sqrt{N}} = \frac{2400}{\sqrt{N}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

For steel gears, machine-cut—

$$\frac{P}{cpb} = \frac{7200}{\sqrt{N}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$

In applying this method of designing gears, whereby first the width is found from equation (84), and this width being substituted in formula (85) gives p, it must be pointed out that the width so obtained will be much too narrow for wheels running at slow speeds. In those cases the width would be taken at not less than $2p$, and this value being inserted in (85), the pitch be found. A preliminary calculation to decide upon a suitable width may, of course, be made, if necessary.

For example, we may calculate the widths and pitch for the two cases cited on p. 156. Case I. To find the breadth we have—

$$b = \frac{PN}{40{,}000e} = \frac{42{,}000 \times 4{\cdot}5}{40{,}000 \times 2} = 2{\cdot}365 \text{ in.}$$

To find the pitch we have

$$p = \frac{P\sqrt{N}}{2400eb} = \frac{42{,}000\sqrt{4{\cdot}5}}{2400 \times 2 \times 2{\cdot}365} = 7{\cdot}9 \text{ in.}$$

This gives a width less than $2\,p$, which is not in accordance with practice. Taking, therefore, $b = 7$

$$p = \frac{42{,}000\sqrt{4{\cdot}5}}{2400 \times 2 \times 7} = 2{\cdot}66 \text{ in.}$$

Case II.

The $\qquad \text{width} = \frac{PN}{100{,}000e} = \frac{85{,}000 \times 9}{100{,}000 \times 2} = 3{\cdot}82$

and the $\qquad \text{pitch } p = \frac{P\sqrt{N}}{7200be} = \frac{85{,}000\sqrt{9}}{7200 \times 3{\cdot}82 \times 2} = 4{\cdot}6 \text{ in.}$

Putting, however, $b = 8\frac{1}{4}$ we obtain—

$$p = \frac{P\sqrt{N}}{7200eb} = \frac{85{,}000\sqrt{9}}{7200 \times 2 \times 8{\cdot}25} = 2{\cdot}14$$

The conclusion is that, in both cases, the teeth might have been narrower and finer in pitch than they were made.

The formulæ (84) and (85) for the design of teeth, viz. $b = \frac{PcN}{80{,}000}$, and $p = \frac{\sqrt{N}}{4800}\frac{P}{b}$, in which e is taken equal to 2, do not, as we see, give good results when the speed of revolution is low. This matter may be further elucidated as follows. We may eliminate P between (84) and (85), writing—

$$P = \frac{80{,}000b}{N} = \frac{4800 \times pb}{\sqrt{N}}$$

or

$$p = \frac{80{,}000}{4800\sqrt{N}} = \frac{16\cdot7}{\sqrt{N}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

From this we see (1) that if N be equal to $16\cdot7^2$, say, 280 revolutions per minute, the pitch will be 1 in., irrespective of the value of P; (2) the pitch will be greater than 1 in. if N be less than 280, no matter how small P may be; (3) the pitch will be finer than 1 in. if N be greater than 280, no matter how great P may be.

The paradoxical result (2) simply means that our formula (84) gives widths of teeth for low speeds which are unsuitably small, and that the pitch must be correspondingly coarse to make up for that. Obviously, the finer the pitch consistent with strength, the more smoothly will the gears run, and, in order to secure the requisite fineness of pitch for low speeds, it will be better to limit the range of speed within which (84) and (85) are applicable without modification.

Let 280 revolutions per minute be the "critical" speed, and let the actual speed in any given example be $N = n \times 280$. Then in (87) we have—

$$p = \frac{16\cdot7}{\sqrt{N}} = \frac{16\cdot7}{\sqrt{n}\sqrt{280}} = \frac{1}{\sqrt{n}} \quad . \quad . \quad . \quad . \quad . \quad (88)$$

Also

$$b = \frac{P \times 280n}{80{,}000} = \frac{Pn}{285} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (89)$$

Therefore

$$\frac{b}{p} = \frac{Pn^{\frac{3}{2}}}{285} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

Now when $\frac{b}{p} = 3$, in equation (90), $P = \frac{855}{n^{\frac{3}{2}}}$

when $\frac{b}{p} < 3$, „ „ $P < \frac{855}{n^{\frac{3}{2}}}$

and when $\frac{b}{p} > 3$, „ „ $P > \frac{855}{n^{\frac{3}{2}}}$

But if N = 280, $n = 1$; therefore

$$\frac{b}{p} \text{ will be } \left\{\begin{matrix}\text{less than,} \\ \text{equal to, or} \\ \text{greater than}\end{matrix}\right\} 3, \text{ according as P is } \left\{\begin{matrix}\text{less than,} \\ \text{equal to, or} \\ \text{greater than}\end{matrix}\right\} 855$$

If, then, in any instance N is less than 280, we shall do well to put $b = 3p$ in the formula $p = \frac{\sqrt{N}}{4800}\frac{P}{b}$, and obtain

$$p = \frac{\sqrt{N}}{4800}\frac{P}{3p}$$

or

$$p^2 = \frac{\sqrt{N}P}{14{,}400} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

as our working formula for determining the value of the pitch. This will always give a suitably fine pitch for gears which have relatively low speeds of revolution, and the width will be three times the pitch.

For steel wheels the formula is—

$$p^2 = \frac{P\sqrt{N}}{21,600} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

We may exemplify the use of the above formulæ by considering the design of *first motion pinions for lathes with cones mounted on the spindle.*

As we have more than once already pointed out, lathes of this type for high-speed or "compromise" purposes ought never to be made of more than 10-in. centres, as otherwise the power they really need can only be supplied by making the belts extravagantly wide. We shall, therefore, limit the calculation to lathes of 6-in., 9-in., and 12-in. centres.

The Load on the Teeth.—We shall take the load on the teeth to be that which will transmit the power required for taking the standard area of cut $\left(\frac{h^2}{6400} \text{ square inches}\right)$ at its proper cutting speed, say, for a compromise lathe, viz. $v = \frac{1}{2a} + 15$—equation (65) of Chapter XVI.

The required cutting horse-power is (*vide* Table XLIII.) $\text{GHP}_{\text{comp.}} = 4 + \frac{h^2}{55}$.

We have for	$h = 6$ in.	9 in.	12 in.
	GHP = 4·7	5·5	6·6

Speed of Revolution of the First Motion Pinion.—We shall take it that the pinion runs, when transmitting the above required horse-power, at the highest speed of the cone, which in this design of lathe is also the highest speed of the spindle. This—by Chapter XVI.—is $N_s = \frac{2750}{h}$;

and for	$h = 6$ in.	9 in.	12 in.
we get	N = 460	306	230 revolutions per minute.

If P is the load on the pinion at its pitch circle of diameter, d inches, we must have—

$$\frac{P \times \pi d N}{12} = 33,000 \text{ GHP}$$

or

$$P = \frac{12 \times 33,000}{\pi} \frac{\text{GHP}}{dN} = 126,050 \times \frac{\text{GHP}}{dN}$$

From practical experience we find—see Fig. 87—that when the spindle is not hollow the relation between size of lathe and diameter of first motion pinion is—

$$d = 2 + \frac{h}{4}$$

so that for	$h =$ 6 in.	9 in.	12 in.
we have	$d =$ $3\frac{1}{2}$ in.	$4\frac{1}{4}$ in.	5 in.
Hence	P = 370	560	725
and, as above,	N = 460	306	230

From formula (84) $b = \frac{\mathrm{PN}}{80{,}000} =$	2·13 in.	2·14 in.	2·08 in.
and from (85) $p = \frac{\sqrt{\mathrm{N}}}{4800}\frac{\mathrm{P}}{b} =$	0·778 in.	0·95 in.	1·14 in.
$h =$	6 in.	9 in.	12 in.
which gives for $\frac{b}{p} =$	2·75	2·25	1·82

More usual proportions would be secured in this case—at all events for the 12-in. lathe—by taking $b = 3p$ and using the formula $p^2 = \frac{\sqrt{\mathrm{N}}\mathrm{P}}{14{,}400}$;

and we should then obtain $p =$	0·75	0·83	0·89
$b =$	2·25	2·5	2·7
$\frac{b}{p} =$	3	3	3

These pinions are intended to be of cast iron, machine-cut, and their proportions may be compared with those plotted on Fig. 87, which are obtained from practice for compromise and high-speed lathes.

Another Method.—We may, on the other hand, decide to make these pinions of equal strength with the belts for compromise lathes as plotted in Fig. 35. We have therefrom—

$$\begin{aligned} \mathrm{B} &= \tfrac{1}{3}(h + 3) \\ &= \tfrac{1}{3}h + 1 \end{aligned}$$

for $h =$	6 in.	9 in.	12 in.
a belt width $\mathrm{B} =$	3 in.	4 in.	5 in.
which at	120	120	120 lb. per in.
gives a pull of	360	480	600
Large cone-step diameter ($2h$)	12 in.	18 in.	24 in.
Pinion diameter $\left(2 + \frac{h}{4}\right)$	3½ in.	4¼ in.	5 in.

The load on the pinion teeth will be greater than that on the belt in the same proportion as the diameter of the largest cone step is greater than that of the first motion pinion. Hence these loads are:—

For $h =$	6 in.	9 in.	12 in.
$\mathrm{P} =$	1235	1920	2880

Now, the speeds at which the pinions transmit these loads are known, being the slowest speed at which the cone runs. These speeds are at once obtained from Table XLVI. by the formula $\frac{\pi \mathrm{A}}{12} n_l = 60\mathrm{V}_l$, where V_l is the belt speed when running on the largest step of the lathe cone. Hence $n_l = \frac{720\ \mathrm{V}}{\pi \mathrm{A}}$ $= 229{\cdot}5\ \frac{\mathrm{V}_l}{\mathrm{A}}$.

Therefore for	$h =$	6 in.	9 in.	12 in.
with	$V_t =$	7	7	7 foot-seconds (two back gears)
and	$A =$	12	18	24
we obtain	$n_t =$	134	89	67 revolutions per minute.

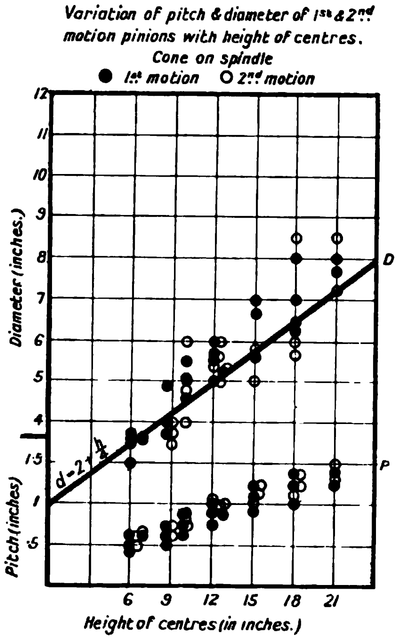

Variation of pitch & diameter of 1st & 2nd motion pinions with height of centres. Cone on spindle
● 1st motion ○ 2nd motion

FIG. 87.

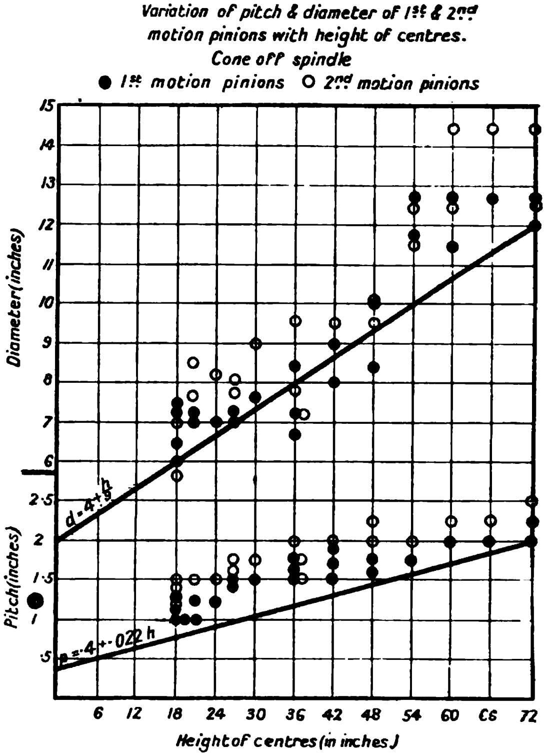

Variation of pitch & diameter of 1st & 2nd motion pinions with height of centres. Cone off spindle
● 1st motion pinions ○ 2nd motion pinions

FIG. 88.

All these speeds are below the "critical" speed (280) discussed above, so that we may preferably use the formula $p^2 = \frac{P\sqrt{N}}{14{,}400}$, which takes the width of the teeth three times their breadth. We thus obtain for the 6-in. lathe—

$$p^2 = \frac{\sqrt{134} \times 1235}{14{,}400} = 0{\cdot}99,\ p = 1 \text{ in.},\ b = 3 \text{ in.}$$

for the 9-in. lathe—

$$p^2 = \frac{\sqrt{89} \times 1920}{14{,}400} = 1{\cdot}26,\ p = 1{\cdot}125,\ b = 3\tfrac{3}{8}$$

and for the 12-in.—

$$p^2 = \frac{\sqrt{67} \times 2880}{14{,}400} = 1{\cdot}635,\ p = 1{\cdot}28,\ b = 3{\cdot}84$$

Tabulating the final results of the two methods, we have—

	For h =	6	9	12	—
Pinions capable of transmitting GHP at the maximum cone speed	p =	¾ in.	⅞ in.	1 in.	Cast iron machine cut.
	b =	2¼	2½	3	
	$\frac{b}{p}$ =	3	3	3	
Pinions capable of transmitting GHP at the lowest cone speed	p =	1 in.	1⅛ in.	1¼ in. +	
	b =	3	3⅜ in.	3⅞	
	$\frac{b}{p}$ =	3	3	3	

Cone off Spindle.—We may next consider lathes in which the cone is mounted on a backshaft, and geared up from the spindle by a pair of wheels of ratio $R_o = \sqrt{h}$ (comp. Chap. XVIII.).

Speed of Rotation.—This may be taken at the highest speed of the cone, *i.e.* $n_g = N_g R_o = \frac{2750}{h}\sqrt{h}$, or $N = \frac{2750}{\sqrt{h}}$. The power to be transmitted is, as before $\text{GHP} = 4 + \frac{h^2}{55}$.

Load on the Teeth.—If d be the diameter of the pinion, we find from practical data that

$$d = 4 + \frac{h}{9} \text{ (\textit{vide} Fig. 88)}$$

then, as before, we must have

$$P = \frac{12 \times 33{,}000\,\text{GHP}}{\pi d N} = 126{,}050\frac{\text{GHP}}{dN}$$

h . . . . . . =	12	18	24	36	48	60	72
$d = 4 + \frac{h}{9}$ . . =	5·3	6	6·7	8	9·3	11	12
$N = \frac{2750}{\sqrt{h}}$ . . . =	792	648	562	459	397	356	324
h . . . . . . =	12	18	24	36	48	60	72
GHP . . . . . =	6·52	9·82	14·4	27·65	46	69·4	98
P . . . . . . =	196	324	483	956	1570	2230	3310
$b = \frac{PN}{80{,}000}$ . . =	in. 1·95	in. 2·6	in. 3·5	in. 5·5	in. 7·8	in. 10	in. 13
$p = \frac{\sqrt{NP}}{4800b}$ . . =	in. 0·59	in. 0·66	in. 0·68	in. 0·78	in. 0·84	in. 0·9	in. 1·0
$\frac{b}{p}$ . . . . . . =	3·3	3·94	5·15	7	9·3	11	13

From this table we see that formulæ (84) and (85) give wheels which are of impracticable width when the loads are high, even although the speeds are above the critical value. It would appear best to use the formulæ by fixing the ratio of $\frac{b}{p}$ preliminarily, say—

$\frac{b}{p} = 5$ for pitches less than 1 in.	$p^2 = \frac{\sqrt{N}P}{24{,}000}$
$= 4$ for pitches from 1 in. to 1·5 in.	$p^2 = \frac{\sqrt{N}P}{19{,}200}$
$= 3$ for pitches from 1 in. to 2 in.	$p^2 = \frac{\sqrt{N}P}{14{,}400}$
$= 2\frac{1}{2}$ for pitches above 2 in.	$p^2 = \frac{\sqrt{N}P}{12{,}000}$

The formula of the form $p^2 = \frac{\sqrt{N}P}{4800b/p}$ then takes values as shown above.

We then obtain the following values for the pitches of the first motion pinions under discussion :—

For lathes with h . . . . =		12	18	24	36	48	60	72
Taking $\frac{b}{p} = 5$	$p =$	0·5	0·6	0·7	0·93	—	—	—
	$b =$	3·0	3·6	4·2	5·4	—	—	—
Taking $\frac{b}{p} = 4$	$p =$	—	—	—	1·04	1·3	1·5	—
	$b =$	—	—	—	4·0	5·2	6·0	—
Taking $\frac{b}{p} = 3$	$p =$	0·63	0·76	0·9	1·2	1·5	1·71	2·04
	$b =$	1·9	2¼	2·7	3·6	4·5	5·1	6·12

If we adhere to $\frac{b}{p} = 3$ throughout—last lines of table—we obtain the linear law $p = 0{\cdot}4 + 0\;0223h$ for the relation between the pitch of the first motion pinion teeth and the height of centres, for lathes with cone off spindle. If we use formula (79), given in Chapter XXIII., viz.

$$p = \left(\frac{p\sqrt{N}}{3000be}\right)^2 = \sqrt[3]{\left(\frac{P\sqrt{N}}{1800}\right)^2}$$

(taking $b = 3p$ and $e = 2$), we obtain the following values of p and b :—

For lathes with h . . . . =	12	18	24	36	48	60	72
,, ,, p . . . . =	0·455	0·595	0·74	1·09	1·45	1·763	2·22
,, ,, b . . . . =	1·365	1·785	2·22	3·27	4·35	5·29	6·66

and from formula (82) we obtain—

For lathes with h . . . . =	12	18	24	36	48	60	72
,, ,, p . . . . =	0·285	0·426	0·58	0·945	1·32	1·64	2·1
$b = (2\frac{1}{2} + p)\sqrt{p}$ . . . . =	1·49	1·92	2·35	3·35	4·4	5·3	6·67
$\frac{b}{p}$ . . . . =	5·24	4·5	4·05	3·55	3·34	3·24	3·18

These results give too fine a pitch for the smaller; and, on the whole, we are disposed to recommend the use of the formula

$$p^2 = \frac{P\sqrt{N}}{14{,}400} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

for obtaining the pitch of a cast-iron machine-cut pinion running at N revolutions per minute, which is to transmit P pounds of load. The width will then be three times the pitch. Some small saving may be secured by increasing the proportion of width to pitch for pitches below 1 in., and a correspondingly greater durability secured; but it may be pointed out that, if a considerable amount of wear is to be allowed before the pinion is discarded, a larger amount of thinning may be permitted if the pitch is initially not too fine.

Second Motion Pinions.—After the proportions of the first motion spur and pinion have been found, it is a simple matter to obtain those for the second motion and subsequent pinions. If P_1, N_1, d_1, and p_1 be the load, revolutions, diameter, and pitch of the first, P_2, N_2, d_2, and p_2 be those of the second motion pinions, then, since—neglecting friction—the work done by each pinion is the same, we have $\pi d_1N_1P_1 = \pi d_2N_2P_2$; and if we take the diameters of both pinions the same, or $d_1 = d_2$, we get

$$P_1N_1 = P_2N_2$$

Let the ratio of the speeds of the first and second motion pinion shafts, *i.e.* $\frac{N_1}{N_2} = g$. Then $P_1N_1 = P_2\frac{N_1}{g}$, or $P_2 = gP_1$.

Now, if we have determined p_1 by the formula recommended above, $p_1^2 = \frac{\sqrt{N_1}P_1}{14{,}400}$ (for $\frac{b}{p} = 3$), we may obtain p_2 at once as follows—

$$p_2^2 = \frac{\sqrt{N_2}P_2}{14{,}400} = \frac{\sqrt{N_1}gP_1}{\sqrt{g} \times 14{,}400} = \frac{\sqrt{g}\sqrt{N_1}P_1}{14{,}400} = \sqrt{g}p_1^2$$

or $$p_2 = p_1\sqrt[4]{g}$$

e.g. lathe of 36-in. centres, cone off spindle, as above, $p_1^2 = \frac{\sqrt{460} \times 956}{14{,}400}$, hence $p_1 = 1{\cdot}2$. Suppose $\frac{N_1}{N_2} = 3 = g$; then $\sqrt[4]{3} = 1{\cdot}32$; so that $p_2 = 1{\cdot}32p_1 = 1{\cdot}32 \times 1{\cdot}2 = 1{\cdot}58$.

The width of the second motion pinion teeth would, of course, be $3 \times 1{\cdot}58 = 4{\cdot}75$ in.

According to Lasche's formula

$$b = \frac{PN}{40{,}000e} = \frac{PN}{80{,}000} \text{ (when } e = 2)$$

we should have $b_1 = b_2$, since $P_1N_1 = P_2N_2$; that is, if all the pinions are to be equally durable, they should all have the same width, and the ratio $s = \frac{b}{p}$ should diminish for the larger and slower-speeded wheels. If it be required to follow this rule, we may proceed as follows:—

Let $s_1 = \frac{b_1}{p_1}$. Then, as before, we find

$$p_1^2 = \frac{\sqrt{N_1}P_1}{4800 s_1}$$

Now $$p_2^2 = \frac{\sqrt{N_2}P_2}{4800 s_2}$$

or $$p_2 = \frac{\sqrt{N_2}P_2}{4800 b_2}$$

but $$b_2 = b_1 = s_1 p_1$$

therefore $$p_2 = \frac{\sqrt{N_2}P_2}{4800 s_1 p_1}$$

or $$p_2 = \frac{\sqrt{N_1}}{4800\sqrt{g}}\,\frac{gP_1}{s_1 p_1} = \frac{\sqrt{g}}{p_1}\,\frac{\sqrt{N_1}P_1}{4800 s_1} = \frac{\sqrt{g}}{p_1}p_1^2 = \sqrt{g}p_1$$

that is, to maintain the same breadth throughout a series of pinions, the pitches must follow the law

$$p_2 = p_1\sqrt{g}$$

e.g. for the 36-in. lathe above we should have, with $s_1 = 3$, $p_1 = 1{\cdot}2$ in. as before, and with $g = \frac{N_1}{N_2} = 3$, $p_2 = \sqrt{3} \times 1{\cdot}2 = 2{\cdot}08$ in. So that we should have for this 36-in. lathe—cone off—

First motion pinion, diameter = 8 in., pitch = 1·2 in., breadth = 3·6 in.
Second motion pinion, diameter = 8 in., pitch = 2·08 in., breadth = 3·6 in.

A very convenient form of diagram for many kinds of lathe calculations is the logarithmic. The diagram, Fig. 89, given below, is specially intended for the calculation of gear ratios, cone-step diameters, countershaft speeds, and in general for the determination of the quantities tabulated in Tables XXVII., XXVIII., and LIII. In the diagram here presented the teeth are designed by our formula $p = \frac{P\sqrt{N}}{4800b}$, which allows for durability as well as strength.

We may first explain the *rationale* of the diagram, and then give an example of its use.

Let it be required to find the geometric ratio r for a given lathe, the values of N_g and N_l and the number of speeds s being given. Then—*vide* Chapter VIII., eq. (35)—we know that

$$S = \frac{N_g}{N_l} = r^{s-1}$$

Then we have $$(s - 1) \log r = \log S$$

or $$\frac{s-1}{1} = \frac{\log S}{\log r}$$

If, now, we have two similar, say right-angled, triangles, ABC and A*bc*—see Fig. 90—it is obvious that $\frac{AC}{Ac} = \frac{AB}{Ab}$; so that, if AC measures $s - 1$ on the same scale as A*c* measures unity, the lengths AB and A*b* will be in the same proportion as log S and log *r*. If, then, we set off a scale of antilogarithms along AX, *i.e.* a scale such that distances from A represent the logarithms of all the numbers from 1 up to the greatest value of $S = \frac{N_g}{N_l}$ we are likely to have, and prepare one or more scales—such as AC—containing as many units as there are likely to be speeds, then by setting the number S on

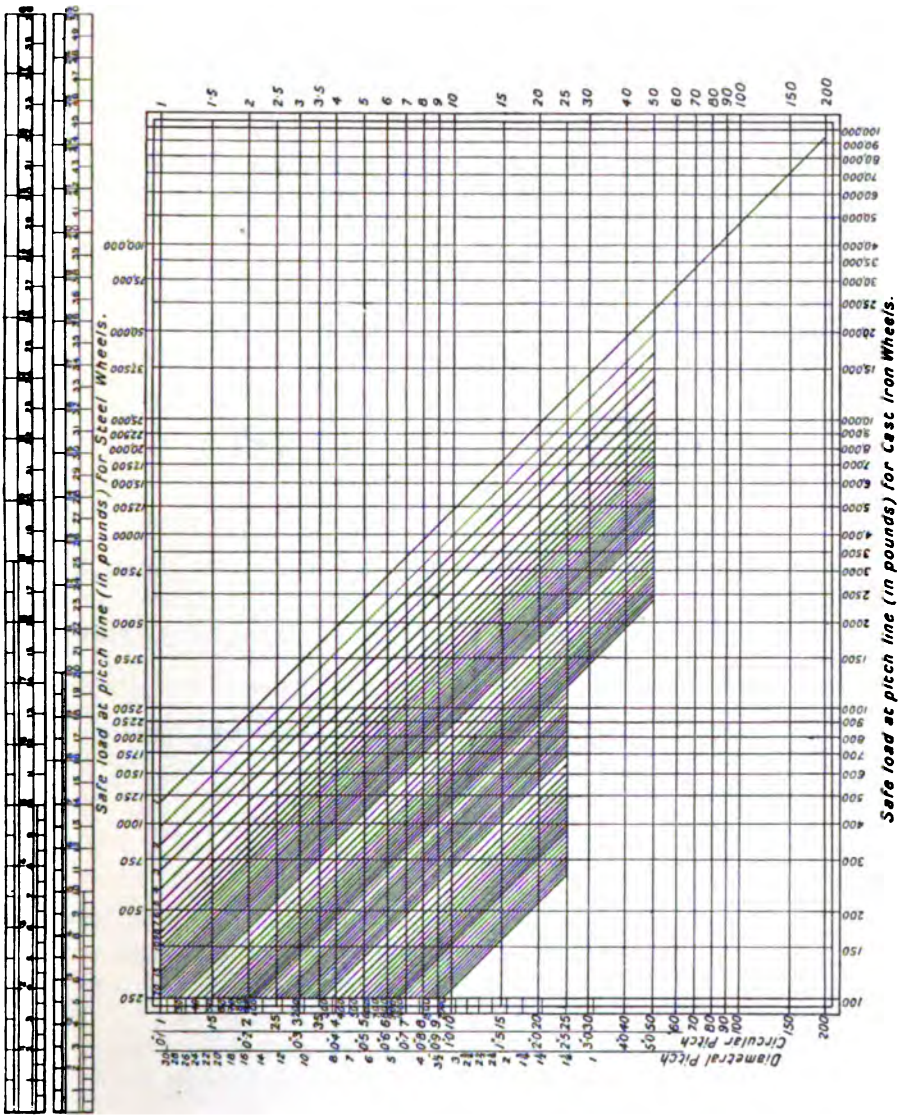


FIG. 89.

the AC scale opposite the vertical BC through B where AB = log S, we shall find A*b* = log *r* by looking opposite *b*, the foot of the perpendicular, to AB, drawn from C, A*c* being unity; *e.g.* (Chapter VIII.) $\frac{N_g}{N_s} = 70$, $s = 32$.

Here we put 31 on AC opposite 70 on AB, and find A*b* = *r* = 1·147.

If the size of the largest cone step is A = 51·4 in., then the size of the smallest is known from $E = \frac{A}{r\sqrt{q-1}}$, where q is the number of steps. Here $q = 4, \therefore E = \frac{51 \cdot 4}{1 \cdot 147^{\frac{3}{2}}}$. If we look opposite A on scale AX, where *de* is the perpendicular through point *d*, A*d* being $= \frac{3}{2}$, we find $1 \cdot 147^{\frac{3}{2}} = 1 \cdot 23$, and $E = \frac{51 \cdot 4}{1 \cdot 23} = 41 \cdot 8$.

Similarly, R_c, the ratio of the two countershaft speeds, is $R_c = r^q = 1 \cdot 147^4 = 1 \cdot 735$, since we find $A_g = 1 \cdot 735$ when $f_g \perp$ to AX through f at 4.

Again, the first back-gear ratio $R_1 = r^{2q} = r^8$, or $R_1 = 1 \cdot 147^8 = 3 \cdot 01$, for A*i* = 3 when A*h* = 8.

Thus, at one setting of scale AC we can determine, with the assistance of the appropriate schedules as given in Chapters VIII. and XXII., all the important gear ratios for any given design.

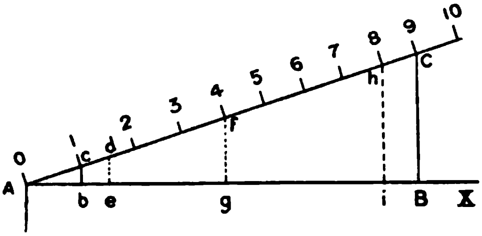


FIG. 90.

Some draughtsmen prefer this method to that by the use of logarithms or the slide rule.

To use the diagram for getting the pitch of the teeth of wheels, as determined from the formula $p = \frac{P\sqrt{N}}{4800b}$, we may first find the pitch which would be required to transmit a given load P at the desired speed N if the breadth were unity.

We then have for p_1 this pitch

$$p_1 = \frac{P\sqrt{N}}{4800}$$

Thus, if the load on a cast iron pinion is 4800 lb. and the revolutions are one per minute, we take the bottom horizontal scale and run up the vertical line through 4800 until we cross the diagonal marked N = 1; the horizontal line through the intersection passes through a pitch equal to 1 in. If the breadth of the wheel is to be 3 in., however, we need only remember that the product of the new pitch p_2 by the new breadth b_2 must be equal to the product of $p_1 \times 1$, since the old breadth was unity. Thus $p_2 = \frac{p}{b_2}$. If, then, we measure down a length on the left-hand vertical scale to the number corresponding to the breadth and set it upwards—in the dividers—from the old value p_1, we get p_2—*e.g.* the new pitch is 0·33; if the breadth b_2 be taken 2 in., p_2 will be 0·5 in., and so on.

CHAPTER XXIV

THE MAIN SPINDLE—DIFFERENT TYPES OF BEARINGS

We now proceed to describe some examples of different types of bearings for lathe spindles, and we may begin with

Bearings for Small Lathes

Figs 91 to 97 show the types of spindle bearings most generally used in lathes up to 12-in. centres. In order that the alignment may not be disturbed by wear, the journals may be made conical, as in Figs. 91 to 95. Fig. 91 shows a very early form, where the bush around the tail journal is keyed to the spindle, but can be freed to slide longitudinally upon it. In this design both journals and bearings were made of steel, hardened and ground between dead centres, then "dollied" into their places. Bearings of this class are to be seen in good condition to-day in lathes which have done hard and continuous work for twenty years. Phosphor bronze has lately taken the place of the steel for the bearings, and the spindles are made of a very hard steel containing about ½ of 1 per cent. of carbon, and having an ultimate tenacity of forty tons per square inch. The front journal in Fig. 91 is tapered towards the tail of the headstock and the rear journal

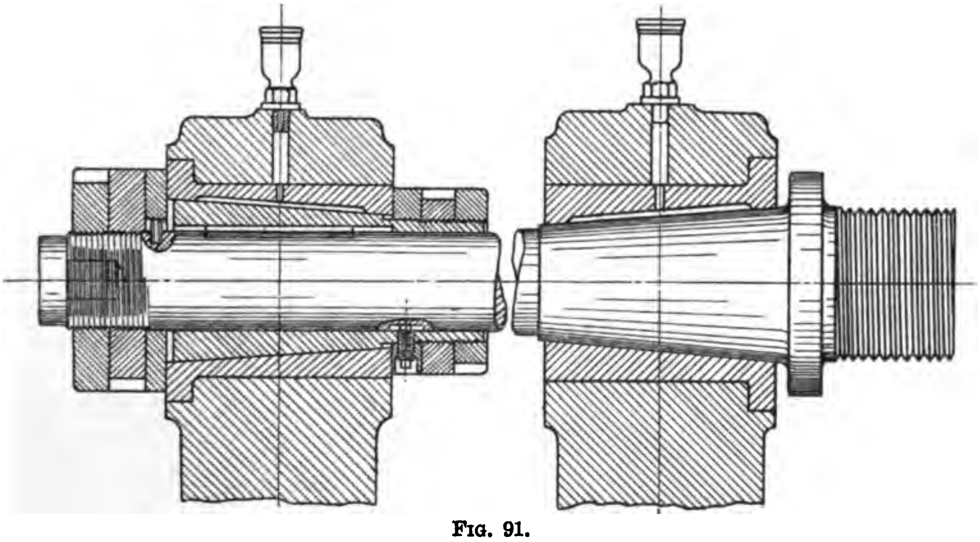


Fig. 91.

in the opposite direction. To allow for wear in the bearings two check-nuts and a washer are fitted to the spindle beyond the tail-bearing, similar provision being made on the end of the tail journal within the cheeks of the headstock. To take up the wear of the front journal the thrust bearing—not shown—is withdrawn; the screw within the washer on the tail journal is freed from the spindle, and the nuts at the tail of the headstock are tightened up until the spindle revolves in the bearing without shake. The tail bearing is then adjusted by the inner nuts on the end of the sliding bush, and the screw within the washer is locked to the spindle to prevent the bush travelling back. Finally, the thrust bearing is readjusted. The wear being more to one side of the bearing than the other, the bush is frequently turned through half a revolution in the headstock cheeks, but should be reground.

Another design of tail bearing of the same class is shown on Fig. 92. Here the lock-nuts and washer within the cheeks and the outer washer have been dispensed with. When the thrust bearing is withdrawn, and the lock nuts on the spindle tightened up, the journals mutually adjust themselves to the bearings. When cutting towards the loose headstock, however, there is nothing to prevent the spindle travelling forward and jamming the tail bush in the bearing. To get over this difficulty, and still maintain the principle of self-adjustment, lock nuts and washer should be fitted to the sliding bush, as in the tail bearing of Fig. 93. Here the spindle is adjusted in the bearings by the nuts on the tail of the spindle; the nuts on the sliding and rotating bush being slacked off whilst doing so, and tightened afterwards to take the pull on the spindle, as when cutting towards the loose headstock, whilst the thrust is taken by the washer and nuts on the spindle, and to the right of the tail cheek.

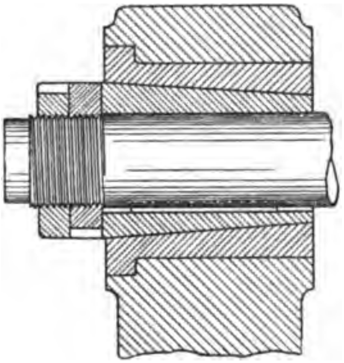


Fig. 92.

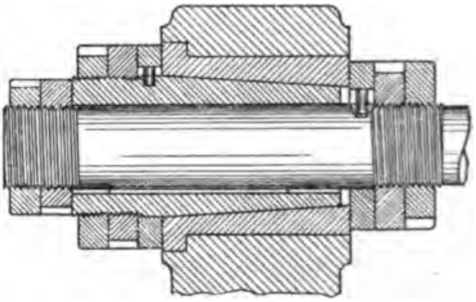


Fig. 93.

The above type of bearing has been considerably displaced by that shown in Figs. 94 and 95. In this case the cones both taper in the same direction, and independent adjustment is provided to each bearing. The bushes can slide laterally within the housings, and are locked by the nuts against the headstock cheeks. The general taper for such bearings is 1 in 6, some makers going as high as 1 in 8, whilst others, to reduce the risk of wedging, make it 1 in 5. This bearing (Figs. 94 and 95) is almost invariably made of phosphor bronze, or of cast iron lined with Babbitt metal. They are then considerably cheaper to manufacture than those previously discussed; and to this fact, no doubt, their adoption is principally, if not entirely, due. It is clear that unless the thrust bearing has been very accurately adjusted the spindle has a double chance of jamming in the bearings; and, in fact, the great objection to conical bearings is their tendency, by reason of the backward thrust on the spindle and improper adjustment of the thrust bearing, to produce excessive bearing pressures with metal-to-metal contact and great friction. Some writers assert that if the taper be made so steep as to prevent jamming the bearing will be too short; that, in fact, the conditions of cool running and moderate bearing pressure are mutually conflicting. This assumes that both diameters are limited. There is no reason, however, why the top diameter should not be the same as the collar beyond the bearing, against which the chuck fits. This arrangement would give a long bearing with a large taper. Another objection raised is that the difference of the rubbing speeds between the top and bottom diameters of the cones produces

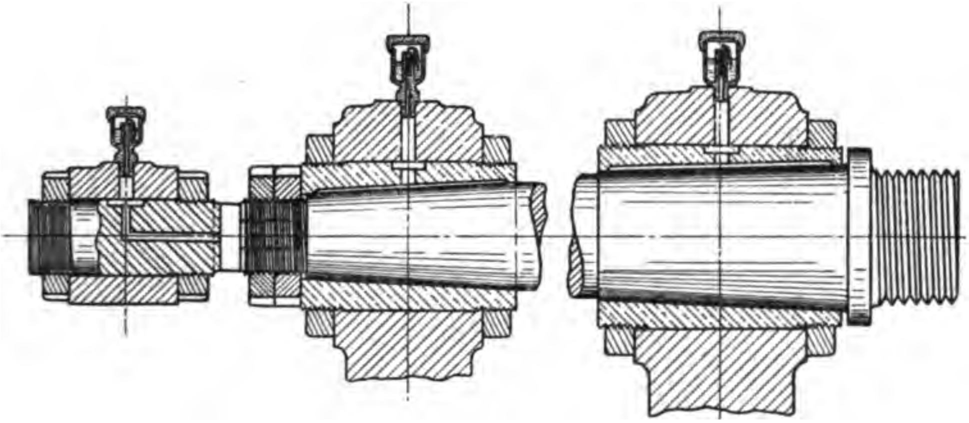


Fig. 94.

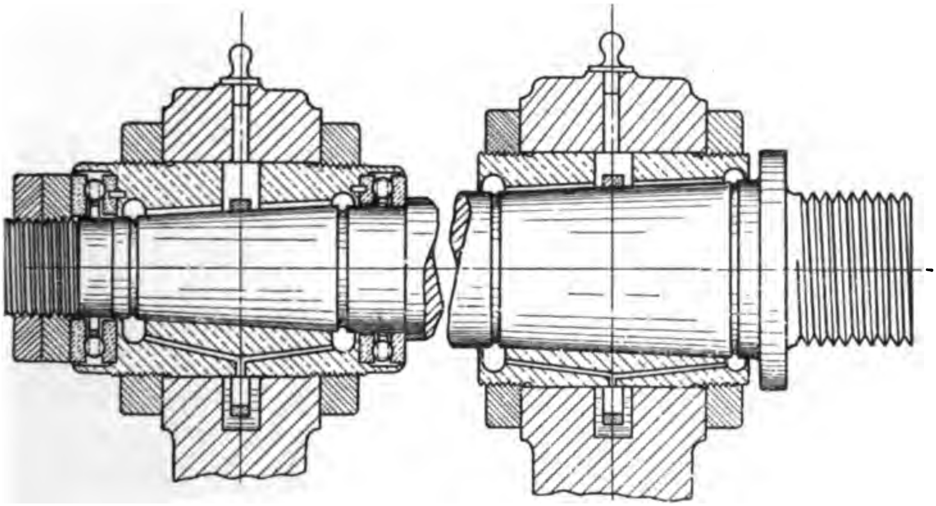


Fig. 95.

unequal wear, and that the wear is greater at the larger diameter. This might be true if the bearing pressure decreased with the diameter; but there is no evidence that it does. Owing to deflection, the pressure may be expected to be greater at the small end, and the wear might then be greater there. An examination of the bearings shows the same wear throughout their length. The above objections to conical bearings would not be heard of if their cost of manufacture could be reduced, and the necessity for careful adjustment dispensed with. The persistence of this type amongst machine tool makers of repute, after thirty years of experience with it, is strong evidence of its general correctness.

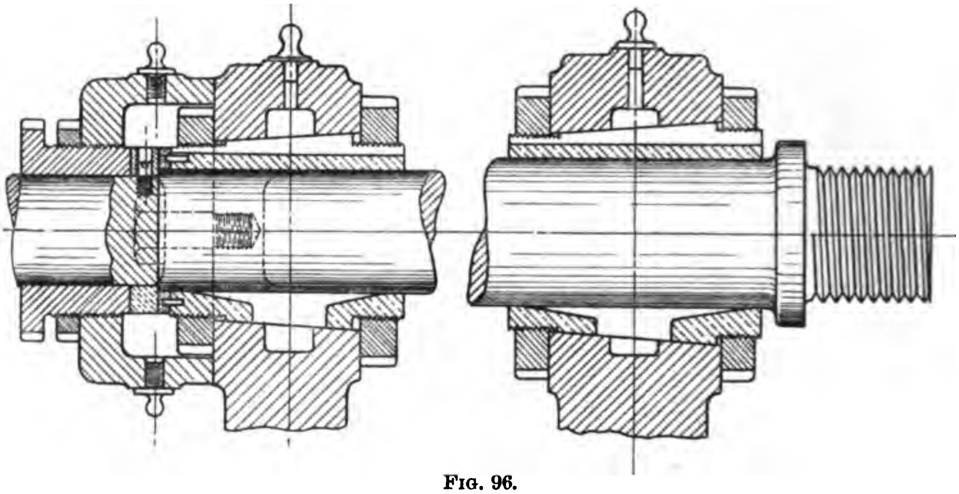


Fig. 96.

In order to obviate these supposed difficulties, many makers have discarded the conical spindles altogether in favour of the designs shown in Figs. 96 and 97. Here the journals are parallel, and the bushes are coned on the outside to fit the housings or bushes in the same. The bushes are split longitudinally right across at one place, and partially at three or four other places, leaving about $\frac{1}{8}$ in. of metal on the inside. Lock nuts are then fitted

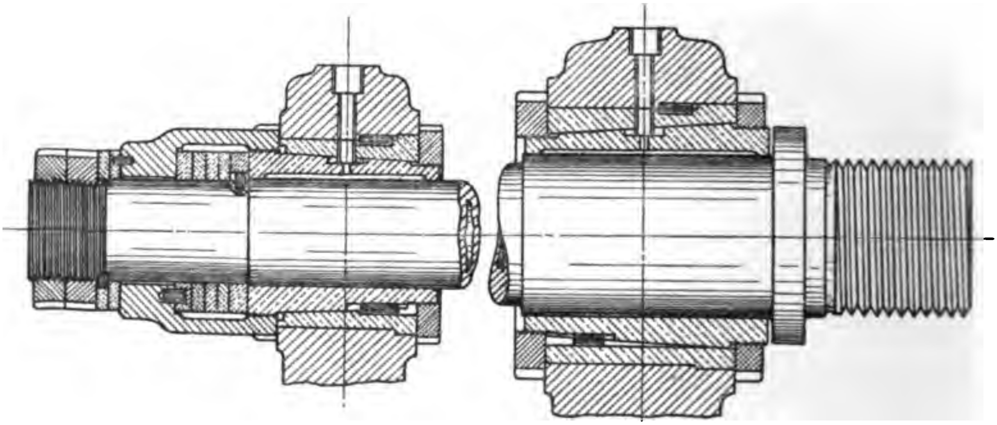


Fig. 97.

on each end, so that when the bush is drawn within the housing and along the cone it closes upon the journal. Each bearing has, therefore, an independent concentric adjustment by the springing of the bushes, and the spindle position is not altered longitudinally.

Although this type is now much used—more than any of the others above described—it has proved incapable of withstanding the wear, and been found almost impossible to secure a good fit without heating after, at most, two or three years' running. This is not surprising when it is understood that the spindle is only supported at several points round its periphery. The springy design also of the bearing makes heavy cutting impossible; in fact, there is considerable chattering when taking a comparatively small cut at a high speed. The spindle has generally been blamed for this weakness, and has been accordingly increased in diameter and fitted with new bushes. For a time it would appear that the chattering has been got rid of, until the bearing becomes worn and requires to be readjusted; then the old trouble begins again, and the same process must be repeated. On the other hand, where the split bush has been replaced by a solid one, not only has the chattering entirely disappeared, but it has been possible to take a heavier cut than before, and the wear has been vastly diminished. Now, the cost of manufacture of the design shown in Figs. 96 and 97 is considerably greater than that shown in Figs. 94 and 95—of which they are modifications—and almost as great as that of the designs of Figs. 91, 92, and 93. Except the need for care in adjustment, no difficulty need be experienced with the use of the conical spindle. With an effective lubricating device it makes a first-class bearing, capable of withstanding heavy cuts at high speeds without undue wear or chatter.

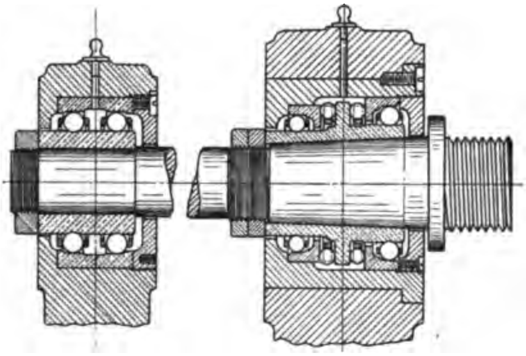


FIG. 98.

Neither the ball nor the roller bearing has been used to any extent for lathe spindles, except for light lathes. The design shown in Fig. 98 is that of the Hoffman Manufacturing Co., which has been occasionally used for this purpose. The front bearing takes the form of the company's well-known, double-compound, ball-bearing. For the front journal the two outer rows of balls form the bearing proper, whilst the two rows in the centre take the thrust and pull of the spindle. The back bearing is the ordinary double-row type, although the makers have found their single-row bearing to give every satisfaction when used here. The lack of bearing surface to give rigidity, the deformation of the balls and plates under heavy loads, and their tendency to scale or crush under very moderate shocks, no doubt accounts for their application being limited to small sizes of lathes. Much the same objections may be advanced against roller bearings, which are difficult to make parallel throughout their length, and are inclined to slew and push back under the load. They are also difficult to adjust for wear.

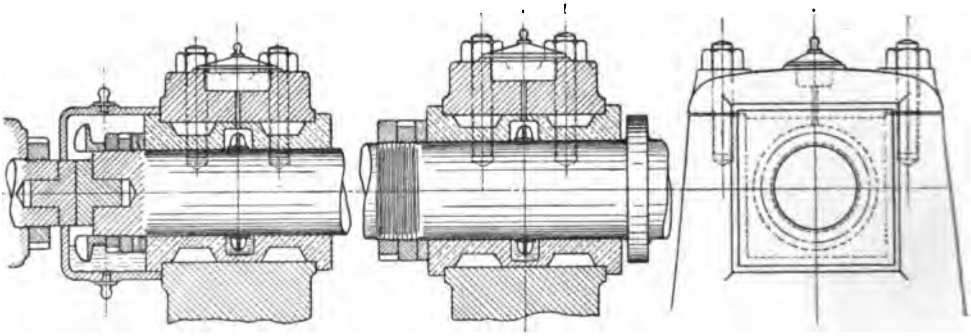


FIG. 99.

Bearings for Heavy Lathes.—For the heavier types of lathes, say from 12-in. centres and upwards, the parallel capped bearings shown in Figs. 99, 100, and 101 are almost universally used. The bearings are in halves, and fit metal-to-metal at the centre of the journal. They are usually made of hard phosphor bronze, cast iron, or either of these lined with white or Babbitt metal. Steel is hardly ever used in this type, and the spindles are rarely, if ever, hardened. The wear is taken up by rescraping the boxes and rebedding the spindle, liners being fitted between the bottom half-box and the headstock, to preserve the height of the spindle. The bearing is centrally placed with reference to the metal of the casting which supports it, the overhang being the same on both sides. It is held thereto by bolts and cast-iron caps, which span the boxes. Fig. 99 shows the cap checked into the

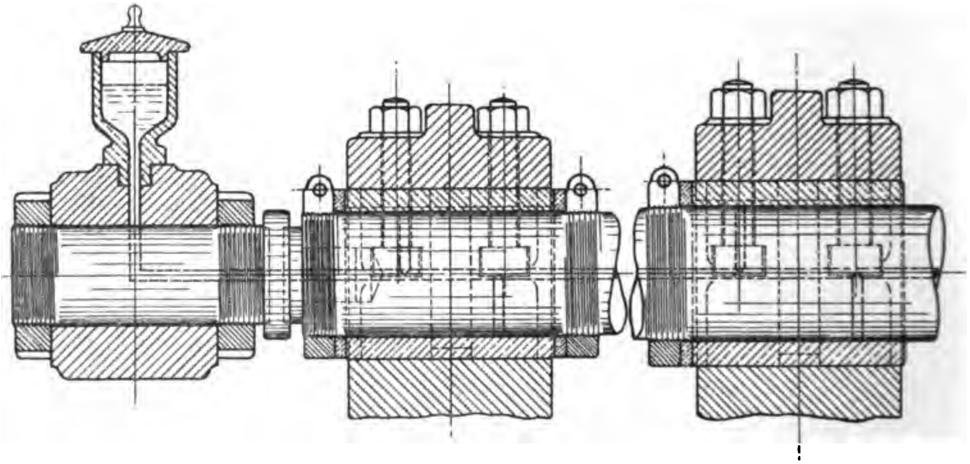


FIG. 100.

head-stock cheek and bearing on the upper half-box, but clear of the cheeks at the top to allow of the cap following the step.

To prevent any endlong movement, the bearing is provided with flanges, which fit on the faces of the cheeks and caps, whereas in the design shown in Figs. 100 and 101 a central bar on both sides of each half-box is used for this purpose. These bars are made to fit the sides of the machined slots in the cheeks. This arrangement is an effective one, and somewhat cheaper than that just described. Messrs. Hulse cast a shelf on the front headstock cheek, on which the flange of the lower half-box rests, to reduce the overhang. With the design of cap shown in Fig. 99 the transverse thrust on the spindle tends to open out the horns of the casting at the bearings.

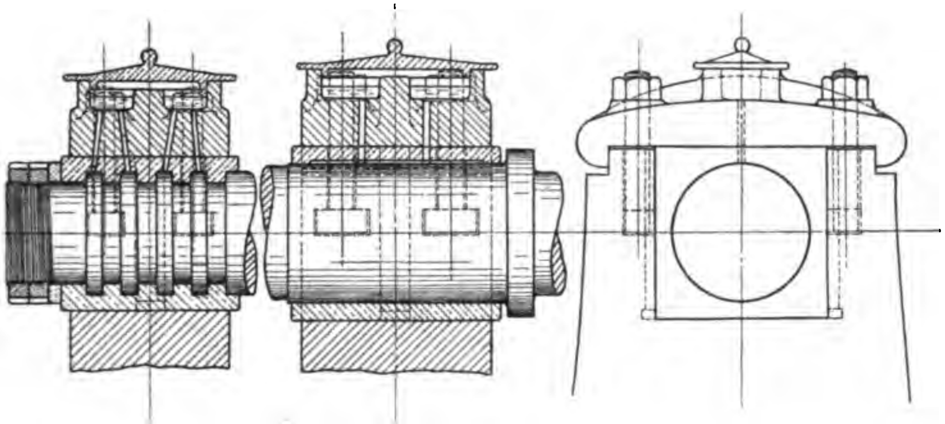


Fig. 101.

When cutting irregular work, headstocks of this type are occasionally broken diagonally across from the lower half-box towards the back. Extra depth will, of course, obviate this; and some makers cast a supplementary buttress at the back. A simpler and more sensible design, however, is that shown in Fig 101, where the cap is made to embrace both cheeks, so that when a thrust comes on the back portion it is communicated through the cap to the front. Thus the one cheek is made to assist the other.

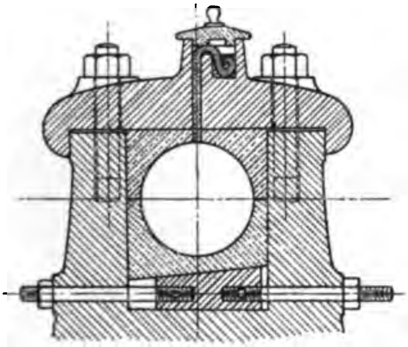


Fig. 102.

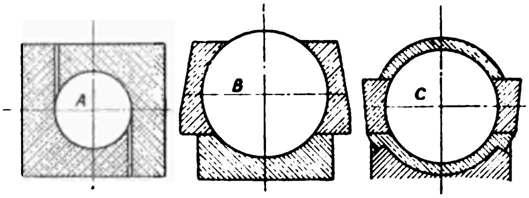


Fig. 103.

Although the capped, parallel-stepped bearings for spindles have had generally no adjustment for wear other than that already described, several American and continental makers have adopted the design shown on Fig. 102.

Here a permanent tapered strip, having a metal-to-metal fit its whole length and adjusted by screws, has been made to take the place of the liners previously referred to. Amongst other styles of steps with wedge adjustments which have found some favour, particularly on the Continent, are those shown at A, B, and C in Fig. 103.

The advantages obtained by making the steps adjustable by means of wedges, screws, etc., for large lathe spindles are questionable, as there is always a tendency to chatter with so many pieces. In the larger sizes of lathes the cone and gear is usually mounted on side shafts, and the spindle is made large to resist bending. It is evident that in such cases a large bearing surface can be obtained by a suitable length of bearing without increasing the length of the headstock, and that the wear can be reduced to a minimum. Thus the fitting of the bearings in halves, accompanied by the periodical examination and overhaul of the steps, may be expected to give the most satisfactory results.

CHAPTER XXV

THE MAIN SPINDLE (*continued*)—SIZING OF BEARINGS FOR STRENGTH AND LUBRICATION

Spindles.—It is usual to calculate the diameter of the front end of the spindle from the twisting moment it is required to transmit; notwithstanding the fact that, in the case of lathes above about 18-in. centres, the twisting moment never is transmitted through the front bearing, but passes round by a pinion into the face-plate wheel.

Following, however, this practice, and taking as the basis of a simple formula, which gives results in agreement with the more recent designs, the twisting moment to be that due to the cutting force corresponding to the standard area of cut acting upon work of a diameter equal to that of the face-plate, we have—compare Chap. VII.—$T = \frac{h^2}{6400} \times 115 \times 2240 \times h = 40h^3$ inch-pounds as this twisting moment, expressed as a function of the size of the lathe.

If d be the diameter of the front journal, and f_s the safe allowable shear stress, we have—

$$T = \frac{d^3 f_s}{5 \cdot 1}$$

or

$$d^3 = \frac{5 \cdot 1 T}{f_s}$$

or

$$d^3 = \frac{5 \cdot 1 \times 40h^3}{f_s}$$

If

$$f_s = 9500$$

then

$$d^3 = \frac{5 \cdot 1 \times 40}{9500} h^3$$

$$d = \sqrt[3]{\frac{1}{46 \cdot 6}} h$$

or

$$d = \frac{h}{3 \cdot 6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

This line is drawn on diagrams Nos. 114 and 115, and is seen to agree very well with results plotted from practice from lathes designed to use carbon steel; but to be smaller than the sizes of current designs for high-speed steel, and "compromise" lathes. The value for these cases is more nearly represented by $d = \frac{h}{3}$

The only reasons which the authors can assign for the use of such heavy spindles—which, as already remarked, are not required to transmit the full twisting moments corresponding to the standard cuts, since the torque passes directly from the face-plate into the work through the dogs—are as follows:—

(1) In face of the unknown stresses to which the use of the high-speed steels gave rise, and which, until the Manchester Committee's experiments were made in 1902–3, were thought to be much greater than those produced by the same cut when taken with carbon steel at the ordinary speeds [some engineers contended the power increased as the cube of the cutting-speed!] toolmakers, in order to be on the safe side, were led to give extravagent sizes to their spindles. A few makers went so far as to make their spindles half the height of lathe centres in diameter; and, a value of one-third the height of centres having been adopted by some, was more or less generally followed by all, as may be inferred from an inspection of our diagram.

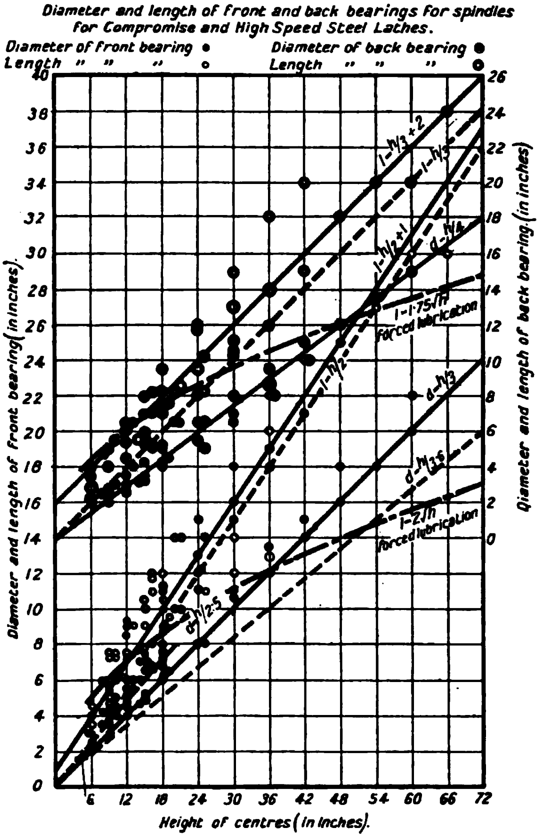


FIG. 104.

There was, of course, no disposition on the part of the purchaser to object to an unduly heavy spindle.

(2) The fatigue of material due to repeated stress and to vibration may be suggested as possible reasons for the employment by makers of the sizes of spindle shown diagrammatically in Fig. 104, and which the authors desire to characterize as excessive.

Now we have throughout maintained that the standard area of cut for a high-speed lathe should be no greater than for a carbon steel lathe, the speed of cutting only being increased.

Under these circumstances the statical stress at high speeds is rather diminished than otherwise—since the cutting stress increases slightly at low speeds—although, of course, the power is much enhanced.

It may well be questioned, on the other hand, whether, with continuous cuts, the uneven finish which frequently accompanies high-speed cutting is responsible for any considerable amount of stress variation in the spindle, the variation of force on the tool being so rapid that only an average value reaches the chuck, at all events with heavy bars of large diameter—and correspondingly large torques. Again, with regard to operating on gapped work, in which the twisted bar, when released from the cutting torque, flies forward, and tends to take the face-plate with it, so overhauling the driving pinion, it may reasonably be argued that at high speeds the bar will not have had time to swing to the other end of its vibration in the short interval which elapses whilst the gap is crossed, whilst at ordinary speeds such a completion of oscillation might have occurred. Thus the torsional stress, instead of being reversed in character, as with carbon steel, may only be reduced in amount whilst remaining of the same nature or sign. In any case, if the torsional stress is transmitted, not through the spindle but by the face-plate, it seems by no means clear in what way such large diameters can contribute to the diminution of torsional vibrations of the work, or to more accurate and

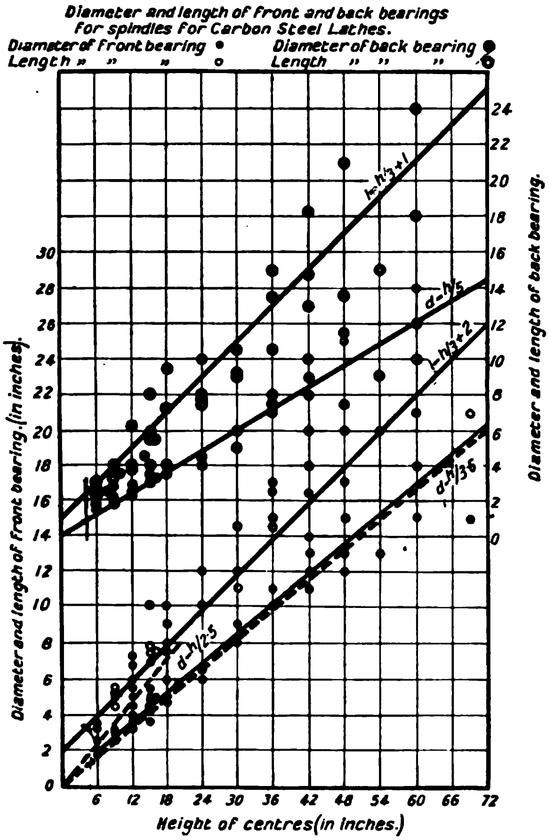


Fig. 105.

efficient cutting. The right place to add weight is in the pinion shaft rather than in the main spindle.

(3) The question naturally arises as to the extent to which bending actions influence the size of the front end of the spindle. It might be supposed that with a large and heavy piece of work overhung from the face-plate the bending moment produced, together with the twisting moment due to cutting, would give rise to so much greater a stress by their combined influence as to necessitate the larger diameters of spindle which appear in practice. To test this, let us suppose a cylindrical job of a diameter equal to that of a face-plate, and a length equal to the height of the lathe centres—which gives about as great a weight as the face-plate dogs ought to be expected to carry—to be overhung and to be operated upon by a tool taking a standard cut at face-plate radius.

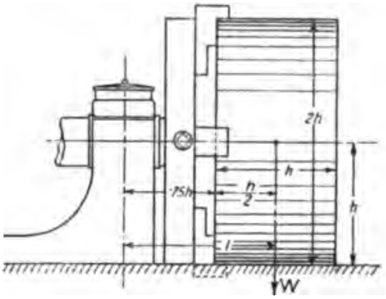


Fig. 106.

Let M be the bending moment,

„ T be the twisting moment,

and let $T_e = 2\sqrt{M^2 + T^2}$ be the twisting moment which would produce the same shear stress in the spindle as the two actual moments give rise to.

Then if f_s is that shear stress, we must have—

$$\frac{d^3 f_s}{5\cdot 1} = T_e = 2\sqrt{M^2 + T^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

Now (Fig. 106) $M = Wl = 0\cdot 26\,\frac{\pi}{4}\,(2h)^2 \,.\, h \times 1\cdot 25h = 1\cdot 02h^4$, or say, $M = h^4$ pounds-inches.

Also, as already calculated above, $T = 40h^3$ pounds-inches.

Take $T = nM$, or $n = \frac{T}{M}$

Then, in this case, $n = \frac{40h^3}{h^4} = \frac{40}{h}$

and $T_e = 2\sqrt{\frac{T^2}{n^2} + T^2} = 2T\sqrt{\frac{1}{n^2} + 1}$

or $$T_e = 2T\sqrt{\left(1 + \frac{h^2}{1600}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

With h . . . . . . =	in. 10	in. 20	in. 40	in. 60	in. 80
$\frac{h^2}{1600} + 1$ . . . . . . =	$1\frac{1}{16}$	$1\frac{1}{4}$	2	$3\frac{1}{4}$	5
and $2\sqrt{\left(\frac{h^2}{1600}+1\right)}$. =	2·06	2·24	2·82	3·6	4·48
d_1 . . . . =	$\sqrt[3]{2\cdot 06d}$	$\sqrt{2\cdot 24d}$	$\sqrt[3]{2\cdot 82d}$	$\sqrt[3]{3\cdot 6d}$	$\sqrt[3]{4\cdot 48d}$
d_1 . . . . =	$\frac{h}{2\cdot 88}$	$\frac{h}{2\cdot 75}$	$\frac{h}{2\cdot 55}$	$\frac{h}{2\cdot 35}$	$\frac{h}{2\cdot 18}$

For twisting only we found above that $d = \frac{h}{3{\cdot}6}$ (94), but when combined bending and twisting are allowed for, we find that d becomes d_1 where d_1 has the values given in the table above.

Now, it is only for the smaller sizes that the spindle is called upon to carry the full twisting moment through the front journal, as for lathes above 15-in. or 18-in. centres the work is almost invariably transmitted into the face-plate wheel directly from a pinion and side shaft. For such (smaller) lathes, then, we may take as a good and safe rule for the diameter of the front journal

$$d = \frac{h}{2{\cdot}5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (97)$$

For the larger sizes, on the other hand, where, according to the annexed table, the spindles would be largely increased by the combination of bending and twisting, since the torque does not pass through the front journal, there seems no reasonable ground for the construction by makers of these very large spindles for the larger sizes of high-speed and "compromise" lathes; and instead of the rule $d = \frac{h}{3}$, which appears to be usual for the diameter of the front journal, we have shown good and sufficient cause for the adoption of the smaller value

$$d = \frac{h}{3{\cdot}6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

This allows a maximum working stress of about 9500 lb. per square inch in the excellent material of which these spindles are made, and it is applicable to both carbon steel, "compromise," and high-speed lathes.

It must be pointed out, however, that this formula applies, of course, to the smallest part of the spindle between the spur wheel and the face-plate. In many small lathes, in which the front bearing has no cap, the spindle is turned down behind that bearing to permit the key of the spur wheel to pass through it. In that case, if δ be this smaller diameter, we ought to take: $\delta = \frac{h}{2{\cdot}5}$ instead of $d = \frac{h}{2{\cdot}5}$.

Let $\delta = cd$, where c is a constant multiplier, giving the ratio of the necked-down diameter of the spindle to that of the front bearing; then we have

$$d = \frac{\delta}{c} = \frac{h}{2{\cdot}5c}$$

Thus if $$c = 0{\cdot}75, \text{ then } d = \frac{h}{1{\cdot}88}$$

If, for example, in an 8-in. lathe we take $d = \frac{h}{1{\cdot}88} = \frac{8}{1{\cdot}88} = 4{\cdot}26$ in., or, say $4\frac{1}{4}$ in. as the diameter of the front journal; then as the spindle is necked down to $3\frac{1}{4}$ in.—*i.e.* to 0·75 of the journal diameter—at the back of the bearing, our rule of $\frac{h}{2{\cdot}5} = \frac{8}{2{\cdot}5} = 3\frac{1}{4}$ still holds good for this smallest part, which means that the shear stress in the spindle is everywhere less than 9500 lb.

Diameter of Back Bearing.—The load to which the back bearing is subjected when the standard cut is being taken will, of course, depend on the position of the tool relatively to the face plate, and upon whether the job is carried between the centres—so that part of the thrust of the cut is taken on the movable headstock—or is overhung. If the tool is assumed to be operating on an overhung job at a distance as far in front of the front bearing as the back bearing is behind it, the load on the latter will be equal to the thrust of the cut, or, say, $P = 40h^2$. If we assume, further, that the journal in the back bearing is equivalent to an overhung circular beam of span $l = (0{\cdot}4 + 0{\cdot}1)h = \frac{h}{2}$, subject to a distributed load of $P = 40h^2$, then its diameter d will be found from $\frac{d_1^3 f}{10{\cdot}2} = \frac{40h^2}{2} \times \frac{h}{2}$. Taking $f = 10{,}000$, we have

$$d_1^3 = \frac{40 \times 10{\cdot}2}{4 \times 10{,}000} h^3 = \frac{h^3}{100}$$

or $$d_1 = \frac{h}{4{\cdot}65} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

The law followed by the diameters plotted in our Figs. 104 and 105, as obtained from experience, is—

$$d_1 = \frac{h}{4} \text{ for high-speed steel lathes} \quad . \quad . \quad . \quad . \quad . \quad (100)$$

$$\text{and } d_1 = \frac{h}{5} \text{ for carbon steel lathes} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

The back journal must be made stiff enough to prevent unequally distributed pressure on the oil film in the bearing. If measurable deflection were to take place the spindle would tend to bear upon the brasses at the bottom inside and top outside edges; the lubricant would be squeezed out and excessive friction and wear would ensue. The deflection of a cantilever of uniform circular section of diameter d—unknown—and of length $l = (0{\cdot}4 + 0{\cdot}1)h$—see above—under a distributed load of $40h^2$ lb. is—

$$y = \frac{Wl^3}{8EI} = \frac{40h^2 \times h^3}{8E \times 0{\cdot}05d^4 \times 8} = \frac{h^5}{2{,}400{,}000d^4}$$

The allowable deflection may be taken to be proportional to the square of the diameter of the journal, *i.e.* $y = nd^2 = \frac{d^2}{1000}$ say. (Compare Osborne Reynolds, "Sci. Papers," II. p. 263.)

Then $$d^6 = \frac{h^5}{2400}$$

$$d = \frac{h^{\frac{5}{6}}}{3{\cdot}66} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (102)$$

The Length of the Front Bearing.—We have given much thought to the consideration of the physical or practical conditions which govern the lengths of the bearings in a lathe spindle, and which fix their lowest limits. The question is one, however, of quite special complexity, and seems to demand much further experimental investigation before being reducible to a strictly rational basis.

In the case of journals such as those of line shafts, pedestals, railway axle-boxes, and the bearings of dynamo electric machines, which run under constant conditions as to load and speed for long periods of time, and in which the direction of the loading forces is never changed, a steady state is reached, after a few hours, in which the temperature of the bearing attains a maximum, and the values of the allowable bearing pressure, the final temperature, and the frictional resistance are known from practical experience or experimental research, and can be predicted for a new design. It is true that the phenomena involved are extremely complex, and we are only now beginning to understand what occurs, consequent upon the researches of Beauchamp Tower and Osborne Reynolds, Stribeck and Lasche, Petroff and Sommerfeld, and many others. We have, nevertheless, fixed rules of design for these simpler cases, and can assign a proper value to the length of a bearing of given diameter when the mode of lubrication has been settled and the conditions of its steady speed and uniform load are known.

The main bearings of a fast headstock are, however, not simply plummer blocks of this kind. In the first place, the load and speed are variable in the highest degree throughout the life of the machine, and the conditions do not—except with the largest lathes—remain constant for a time long enough to enable the above described steady state to be attained. Thus, if these bearings were designed as ordinary pedestals for the highest speeds and the maximum loads to which they are to be subjected, as if for the steady state, they might be expected to be much too large, wasting both power and material. Contrariwise the journals of a lathe spindle have to run for a large proportion of their lifetime at excessively slow speeds under moderate or even great loads; and under these circumstances the lubricant will usually be squeezed out, so allowing the journal and brass to rub metal against metal with considerable friction and great wear. Again, there is the condition of the maintenance of accurate axial alignment, which is in this case of paramount importance; and we have to decide whether it is better to attempt to maintain by forced lubrication under all circumstances a complete oil film all round the journal in order to prevent wear altogether; or, on the other hand, to permit the metallic contact and wear to take place consequent upon a less perfectly controlled supply of lubricant, whilst providing mechanical means for readjusting the alignment with ease and certainty.

In order to discuss the question thoroughly in all its aspects, it will be necessary to go into a somewhat lengthy description of the phenomena accompanying the rotation of a journal in a bearing, from which it is either partially or completely separated by a supply of lubricant.

A. **Dry friction.**—Commencing with the case of the spindle lying in its bearings at rest, and then gradually attaining any given speed, we find that the brass, being somewhat larger in bore than the diameter of the journal, is touched by the latter only along a narrow line at its lowest point. When rotation commences, the journal rolls up towards the front of the lathe until the load on the bearing—supposed vertical—makes an angle equal to the maximum angle of repose with the journal radius drawn to the point of contact. The point of contact K—see Fig. 107—is then in a plane distant

$r \sin \phi$ from that containing the centre of the lathe, and the journal slides on this point, or rather line, with the three forces f, n, and W in equilibrium. The law followed is that of so-called "dry friction," and the coefficient of friction $\mu = \tan \phi = \frac{f}{n}$ is identical with that found by Morin in 1833 for slightly greasy metallic surfaces, viz. from 0·12 to 0·18—or say, 0·15 on the average.[1] The ratio $\frac{f}{W}$ is, of course, equal to $\sin \phi$, and being nearly the same as $\tan \phi$ for small angles—such as 8½ deg.—we write for the frictional work done per revolution—

$$\text{Friction work} = \frac{2\pi r f}{12} = \frac{\pi d \mu W}{12} \text{ ft.-lb. per rev.}$$

If $d = \frac{h}{3}$, $\mu = 0{\cdot}15$, and $W = 3h^3$ lb.—being half the weight of a job equivalent to the lathe's volumetric capacity—we have

$$\text{Friction work} = \frac{0{\cdot}15\pi}{12}\,\frac{h}{3}\,3h^3 = 0{\cdot}0393h^4 \text{ ft.-lb. per rev.}$$

Now the cutting work is:—

$$F\frac{\pi \times 2h}{12} = \frac{2240 \times 100h^2}{6400}\,\frac{\pi h}{6} = \frac{35\pi}{6}h^3 = 18{\cdot}3h^3 \text{ ft.-lb. per rev.}$$

Thus, the ratio of the work spent on friction to the work spent upon cutting is

$$\frac{0{\cdot}04h^4}{18{\cdot}3h^3} = 0{\cdot}0022h$$

a quantity increasing proportionately with the size of the lathe. For a 10-in. lathe, *e.g.*, the work wasted is 2·2 per cent., for a 50-in. lathe it is 11 per cent. of that usefully employed.

If, however, the thrust on the bearing is due rather to the cutting force than to the weight carried in the centre—as is usually the case in small lathes—the point of contact will be at the back of the bush in the upper quadrant at k—see Fig. 108—obtained by drawing the resultant (P) of the cutting force (F) and the weight (W) a tangent to the friction circle of radius $r \sin \phi$.

If, now, the journal keeps sliding past the point k, and the friction coefficient remains 0·15—no oil film entering to separate the surfaces—it is clear that the efficiency is the same for all sizes of lathe. For

$$\text{friction work} = \frac{2\pi r \mu P}{12} \text{ ft.-lb. per rev.} = \frac{\pi h \times 0{\cdot}15P}{36} \text{ if } d = \frac{h}{3},$$

whilst cutting work is, as before, 18 $3h^3$ ft.-lb. per rev.

---

[1] See, *e.g.*, Moseley's "Principles of Engineering," 1843, pages 149–155.

Taking, for simplicity

$$P = F = \frac{224{,}000h^2}{6400} = 35h^2$$

the friction work is

$$\frac{0{\cdot}15\pi \times 35h^3}{36} \text{ and } \eta = \frac{0{\cdot}1666 - 0{\cdot}004}{0{\cdot}1666} = 0{\cdot}975$$

or 2½ per cent. of the work is lost by friction of the front journal when the speed is so slow that no oil separates it from its bearing.

B. **Resistance due to Lubricant.**—Returning to our former case, where the weight of the job forms the predominant part of the load on the journal, and the point of contact is at first at K—Fig. 107—near the bottom, there comes a time as the spindle speed goes on increasing when the oil is carried round with the journal, and forms a film separating it completely from the bush.

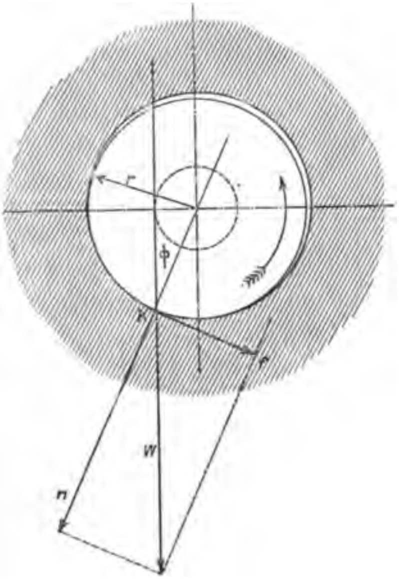


FIG. 107.

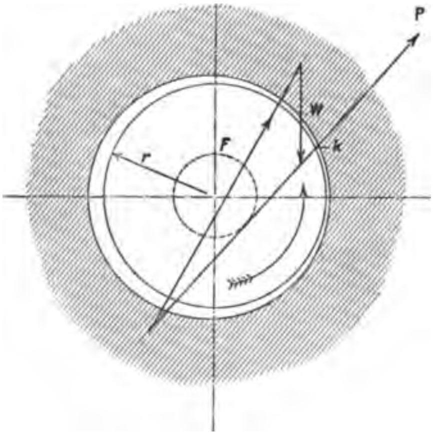


FIG. 108.

The point of nearest approach between journal and brass is by this means shifted round to B, at the back of the bush, and in the same horizontal plane as the centre—see Fig. 109.

The distribution of pressure all round the journal—the interspace being all filled with lubricant—is also shown diagrammatically in Fig. 109.

When the speed of rotation is slow, the pressure is very great at p_1, just before B, the point of nearest approach, is reached, and falls to nothing at p_2 just after it is passed. It then soon rises and maintains a nearly constant value right round from C through A to D, after which it quickly increases to its maximum.

Sommerfeld has given an easy proof that, for the journal to remain in equilibrium under the forces due to the frictional and fluid pressure of the oil film, the load must act perpendicularly to the line joining the centres of the bush and the excentrically situated journal, and on that side of the centres away from the point of nearest approach. This is indicated by the load line marked W—Fig. 109.

For greater values of the speed of rotation the maximum value of the oil pressure diminishes, and its position (q_1) recedes from the neighbourhood of B; whilst at the same time, the position of the place where the oil pressure is nothing (q_2) advances by an equal amount on the other side of C. The point of nearest approach of journal and brass still remains at B, however; but the journal becomes less excentric. If the speed could be made infinitely great the journal would run centrally in the brass, and the position of maximum and minimum fluid pressure would be vertically below and above the centre, viz. at s_1 and s_2—Fig. 109.

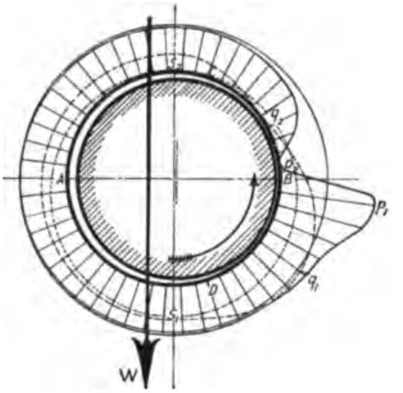


FIG. 109.

If the original difference of radii of bearing and journal were known, as also the viscosity of the lubricant, Sommerfeld has shown how the actual value of the maximum fluid pressure and its circumferential position, as also the amount of the excentricity of the journal in the bearing, could be calculated. The above difference would be very difficult to find, however, and, even if it were known, the unknown rise of temperature due to the friction of running would soon vitiate the figures, owing not only to the differing amounts of expansion of the journal and the bush, but also to the varying values of the viscosity of the oil within the film itself. (Comp. Osborne Reynolds, "Sci. Papers," vol. ii., pp. 282, *et seq.*)

We are unable, therefore, to give quantitative results; but from qualitative reasoning, we see that the pressure under which the oil must be supplied in order to keep up a complete film is greater for low speeds of rotation than for high, and that the positions of maximum and minimum oil pressure vary with the speed, and, of course, the direction of the load on the journal.

Lubrication Automatic.—If the oil is fed in by an ordinary cup and syphon, or by a ring or centrifugal method of supply, it should obviously be made to flow on to the journal at the place where the pressure is least; *i.e.* at p_2, q_2, or s_2, Fig. 109 (*ante*). It will not enter at p_1 or q_1 unless it is under a greater head than that of the oil film at these points.

The oil ways should therefore be fed from a point situated in the top rear quadrant of the bearing when the journal is loaded by gravity only, and the point should be further back the slower the speed. This applies, then, especially to large lathes.

If the loading of the journal is principally due to cutting force acting upwards upon it, the feed should obviously be placed in the bottom front quadrant, and nearer the front the slower the speed of rotation. This meets the case of the smaller-sized lathes.

The compromise ordinarily effected to allow the lubricant to enter, whatever may be the direction of the loading, is the simple one of fitting the oil cup on the top of the bearing. This seems almost the only thing to do in the case of automatic lubrication; but it is the correct position only when the resultant force upon the journal, due to gravity and cutting force, etc., acts nearly horizontally and from front to rear.

Forced Lubrication.—When the lubricant is supplied by mechanical means at a fixed rate and at any required pressure it will preferably be fed in at the points (p_1, q_1) of greatest oil pressure in the bearing, although for engine bearings it is usually fed in at the points p_2, q_2, and s_2.

For large lathes, where gravity is more important, this region of greatest pressure lies in the rear bottom quadrant.

For small lathes, on the other hand, in which the force on the spindle acts upwards, owing to the cutting force being relatively greater, the maximum oil pressures occur in the front top quadrant.

To meet all contingencies it would appear, on the whole, best either to force the oil in at the back of the bearing, well below the centre; or, preferably, to fit three alternative branches from the oil pressure supply pipe to the back, top, and front, any one of which may be turned on at will to suit the conditions of working.

THEORY OF LUBRICATION

In describing the phenomena occurring when a journal rotates in a bearing, we have, so far, not alluded to the nature or magnitude of the frictional resistance experienced when there is an abundant supply of lubricant completely separating the former from the latter, and preventing any metal-to-metal contact.

It is frequently stated that "there is no friction without abrasion," or, in other words, that unless two substances rub against each other, there can be no resistance due to relative motion. This, however, is not the case. When a film of lubricant is interposed between two metallic surfaces there is a resistance to relative motion of these surfaces due to the shearing or transverse distortion of the oil film.

In the case of the shearing of a solid, we know that Hooke's law, *ut tensio sic vis*, applies; so that if q, Fig. 110, be the shear stress, and $\frac{s}{y}$ the shear strain, we write $q = G\frac{s}{y}$, where G is the modulus of transverse elasticity. But if the surfaces AA, BB are separated from one another by a liquid of thickness y, and AA moves with speed v relatively to BB, the resistance no longer depends on the mere magnitude of the angle θ, but upon the rate of its increase with time. If a is the area of either surface, the force required to maintain the steady speed v is

$$F = qa = \frac{kav}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (103)$$

where k is called the coefficient of viscosity. [This coefficient is clearly that force which will move a unit area of surface with unit speed relatively to another surface from which it is separated by a liquid of unit thickness.]

A film of oil of *uniform thickness* y, all round a journal of diameter d and length l, and adhering to it and to the bearing surrounding it, will thus oppose a resistance to the rotation of the former, with surface speed v, of the amount

$$F = \frac{k\pi dlv}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (104)$$

In such a case the *friction would not depend on the load*, which does not appear at all in the expression; and it will be readily admitted that the force required to distort the oil film cannot depend on the fluid pressure to which it is, at the moment, subjected. The friction is governed only by the area of viscous fluid to be sheared, and the viscosity of the oil—*i.e.* the kind of oil and its temperature (with which the viscosity greatly alters)—and it also gets greater the smaller (y) the thickness of the film, so that if the journal is a close fit within its bush the resistance to motion will be greater than if the fit is an easy one.

There are, unfortunately, very few cases in engineering practice where a journal rotates with a uniform thickness of oil around it. It is only when the speed is very great indeed that this can happen—as with steam turbines and mill spindles. At moderate and low speeds we know, as above described, that the shaft moves to one side by an amount which depends on the speed and the load; the excentricity for any given load becoming greater the smaller the speed. We have just seen that the frictional resistance depends on the thickness of the oil film, but it is known from the researches of Reynolds and Sommerfeld that the diminution of resistance, due to the thickening of the film on one side of the shaft, is more than made up for by the increase due to the thinning on the other; so that, on the whole, the friction gets greater as the journal becomes more excentric.

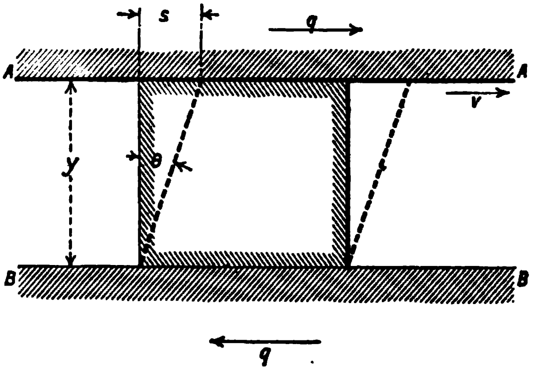


FIG. 110.

Starting, therefore, with a bearing in which the shaft is running at a very high speed, and is nearly concentric with its bush, the friction should diminish directly as the speed, if the simple law of resistance to oil-shearing were followed; since, however, as the speed gets less the excentricity increases, and on this account the friction gets greater, we shall find that at low speeds, when the excentricity becomes very marked, the increase due to this more than makes up for the diminution due to lessening speed, and the total resistance attains a minimum value at a certain speed—which depends upon the pressure. This has been very well worked out by Sommerfeld, whose curves of friction and speed, deduced from theory, show a satisfactory agreement with experimental results, *so long as the speeds of rubbing are less than about* 20 *ft. per minute.* For speeds greater than this the change of excentricity is not so important, and the law of fluid friction is interfered with for another reason. *On account of the high speed the temperature of the oil film rises, and its viscosity is thereby reduced. The friction then increases less rapidly than in proportion to the speed.*

At such high speeds as 1800 ft. per minute, Lasche, indeed, found that the friction was quite independent of speed. We do not meet with such rubbing speeds in the bearings of a lathe, however, and we need not here refer further to this point.

The curves of frictional resistance obtained by experiment are shown in the annexed figure (111). They are principally those found by Stribeck for experiments with a Sellers bearing (*v.* Z. d. V., September 6, 1902) having double-ring lubrication, the material being gun-metal.

The two thick dotted curves in this figure are taken from Heimann's report on experiments by the method of Dettmar (*v.* Z. d. V., Bd. 49, p. 1161).

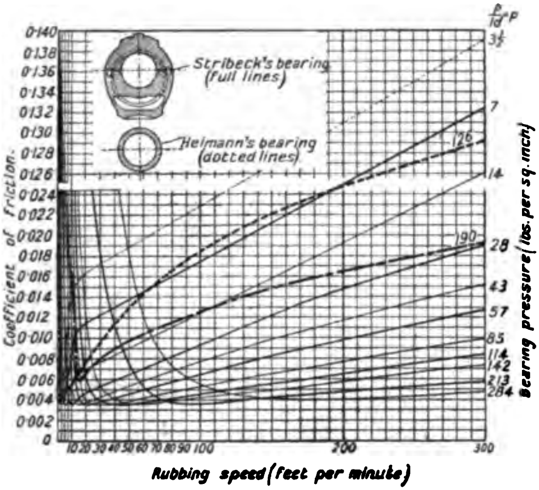


FIG. 111.

They refer to a *solid bush* of gun-metal, with an unbroken film of lubricant all round, and are, therefore, perhaps more directly applicable to the case of a lathe spindle than those of Stribeck.

The curves give the "coefficient of friction" on a base of speed of rubbing in feet per minute.

The term "coefficient of friction" denotes the ratio of the total resistance, due to the shearing of the lubricant, to the load on the journal; and although it has no physical meaning, it supplies a convenient and conventional method of discussing the results.

If F denote the resistance, and P the load, we have (from equation 104)

$$\mu = \frac{\mathrm{F}}{\mathrm{P}} = \frac{k\pi dlv}{\mathrm{P}y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

and if $\frac{\mathrm{P}}{dl}$ be, as usual, denoted by p, and called the *bearing pressure*, or load per unit of projected bearing area, we see that

$$\mu = \frac{k\pi}{y}\frac{v}{p} = c\frac{v}{p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (106)$$

or the *coefficient* of friction would vary as the speed and inversely as the bearing pressure *if the oil film were of uniform thickness all round, and the temperature of the lubricant were constant.*

Referring to the dotted curves—thick lines (Fig. 111)—we see that μ is not proportional to the speed in the actual experiments even when the temperature of the *bearing* is kept constant throughout.

It is found that μ increases less rapidly than the velocity of rubbing when the speed is greater than about 20 ft. per minute. This is due to the fact that the difference between the actual temperature of the oil film and that of the bearing gets greater as the speed increases, so that although the *bearing* temperature may remain the same throughout a series of experiments, that of the film of lubricant itself, upon which the value of k—the coefficient of viscosity—depends, will rise. On this account μ is found to increase more nearly as the square root of the speed as the rubbing speed becomes larger.

These—as also Beauchamp Tower's—experiments show that the "coefficient of friction" in a lubricated bearing follows the law $\mu = \frac{c\sqrt{v}}{p}$ (instead of $\mu = \frac{cv}{p}$; equa. 106) for the speeds which are usual in lathe spindles—viz. from 20 ft. to 300 ft. per minute, and for the bearing pressures commonly occurring.

According to the same equation (106) the constant depends upon the thickness (y) of the oil film—as to this see below—and upon the coefficient of viscosity (k).

Now we have found that for machinery oil, a simple expression gives the values of k with fair correctness between the temperatures 70° and 160° Fah.; the more correct expression of Osborne Reynolds being—$k = 0{\cdot}000047e^{-0{\cdot}0231t}$.

Our formula is $k = \frac{\mathrm{B}}{(t - 60^\circ)}$, or $k = \frac{\mathrm{B}}{\theta}$, where B has the value $\frac{1}{500}$.

Thus we write
$$k = \frac{1}{500\theta}. \quad \ldots \ldots \ldots \quad (107)$$

A roughly accurate expression for the coefficient of viscosity for machinery oil as deduced from these experiments will therefore be (for speeds over 20 ft. per minute)

$$\mu = \frac{\mathrm{C}\sqrt{v}}{\theta p} \quad \ldots \ldots \ldots \quad (108)$$

or

$$\mu = \frac{900\sqrt{v}}{500\theta p} = \frac{1{\cdot}8\sqrt{v}}{\theta p} \quad \ldots \ldots \quad (109)$$

Here v = surface speed in feet per minute,
p = bearing pressure in pounds per square inch,
and θ = excess of oil temperature above 60° Fah.

Upon further examination of the experimental results plotted in curves upon Figs. 111 and 112, we observe that for speeds just above the speed of minimum friction the coefficient of friction diminishes for a little in exact proportion to the speed, but that (as the minimum point is approached) the effect of the relatively great excentricity of the journal in the bush begins to have an important influence in increasing the friction; and the coefficient attains a minimum value as prescribed by Sommerfeld, and described above.

As to the positions of the minimum points, we see that the coefficient of friction takes its least value for speeds which depend upon the bearing pressure; those speeds being greater the greater the bearing pressures—Fig. 111.

After the minimum points are passed the curves begin to rise more or less rapidly as the speed continues to fall. This is due to the oil film being squeezed out of the bearing; and we shall find it necessary to attach great importance in the sequel to this so-called "minimum point," as bearings must be designed so as to avoid such speeds and pressures as correspond to them.

In Fig. 112, reproduced from Heimann's article (*loc. cit.*), we observe the effect on the coefficient of friction and on the positions of its minimum points, of an increase of the *temperature of the bearing.* The higher the temperatures at which the bearing runs the smaller the coefficient of friction, and the greater are the speeds at which the minimum values of that coefficient are attained.

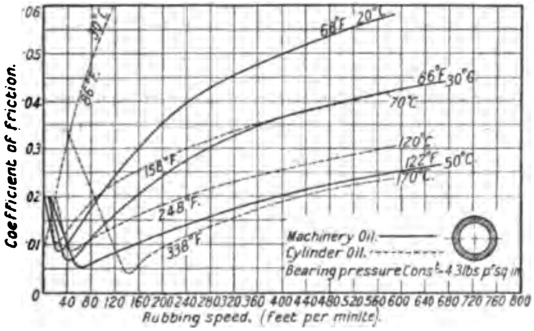


FIG. 112.

In conclusion, we observe that the curves all rise very rapidly when the speed is very slow and the oil has all been squeezed out of the bearing; and that they tend to converge upon the constant value 0·15, the coefficient of friction for greasy metals sliding upon each other, long ago determined by Morin. It is only then that μ remains constant, and that we encounter a frictional resistance and loss of work proportional to the load on the bearing, and of the amount already calculated.

C. **Application to Spindle Bearings.**—In endeavouring to apply the theoretical explanations and experimental results which have been discussed in the last two articles to the proportioning of the bearings for a lathe spindle we have two extreme conditions to provide for.

First, the bearings must keep cool when the spindle is running at its *highest* speed of revolution and under the heaviest load it will be called upon to sustain, *e.g.* when carrying a heavy job between centres and running at polishing speed.

Second, when taking a heavy cut, or carrying a heavy piece of work at a *slow* speed of rotation, it ought to be possible to prevent the metallic contact of journal and brass from taking place, which has been above described, by always retaining a film of oil between the same at every part of the circumference.

With reference to the *first case,* we have to consider how much heat is generated by the viscous resistance opposed by the oil to the rotation of the shaft, in what manner this heat is dissipated, to what temperature the bearing will rise in actual running, and what law connects the temperature at which the journal may safely run with the pressure and the speed.

The heat generated is obviously equal to $\frac{\mu P \pi d N}{12J}$ thermal units per minute, if d is the journal diameter in inches, P the load in pounds, μ the coefficient of friction, N the number of revolutions per minute, and J. Joule's equivalent.

If the heat dissipated by radiation from the surfaces of the housing and shaft be, as usually assumed, proportional to dl, the projected bearing area, and to θ the temperature difference between bearing and external air; then, when the temperature of the bearing becomes steady, we shall have (e being the emissivity coefficient)

$$\frac{\mu P \pi d N}{12J} = edl\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

In the standard text-books—*e.g.* Unwin's "Machine Design," vol. i. p. 240—it is usually assumed *that whatever may be the diameter or length of the proposed journal the value of μ will remain constant.* In that case

$$l = \frac{\mu P \pi N}{12Je\theta} = \frac{\mu\pi}{12Je\theta}PN = \frac{PN}{\beta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (111)$$

which means that the length of the bearing must increase with the load and the revolutions.

As a matter of experimental fact, however, μ is by no means constant. It depends on speed, pressure, and temperature, and, as we have already seen, we may take it to follow the law

$$\mu = \frac{1{\cdot}8\sqrt{v}}{\theta p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (112)$$

If, then, we insert this value in (110), we obtain

since $$v = \frac{\pi d N}{12}$$

therefore $$\frac{cP(\pi d N)^{\frac{3}{2}}}{12^{\frac{3}{2}}p\theta J} = edl\theta$$

With $c = 1{\cdot}8$, $\frac{P}{p} = ld$, and $J = 778$, we get

$$\theta^2 = \frac{(dN)^{\frac{3}{2}}}{3230e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (113)$$

To determine the value of e, the emissivity coefficient, we may apply this formula to Stribeck's results from his Sellers bearing (see his figure 5, Z. d. V., September 6, 1902). This bearing, of diameter 2·76 in., attained—according to that figure—after running for three hours at 760 revolutions per minute, a final temperature of about 140° Fah.

Therefore $\theta = 80°$ Fah.

Hence from $e = \dfrac{(dN)^{\frac{3}{2}}}{3230\theta^2}$

we obtain $e = \dfrac{(2{\cdot}75 \times 760)^{\frac{3}{2}}}{3230 \times 80^2} = \frac{1}{220}$ B. Th. U. per min. (114)

dissipated per square inch of projected bearing area per degree difference of temperature Fah., between the bearing and the outer air.

Thus the formula for θ_f, the final rise of temperature of a bearing above its surroundings, becomes

$$\theta_f = \frac{(dN)^{\frac{3}{4}}}{\sqrt{3230 \times \frac{1}{220}}} = \frac{(dN)^{\frac{3}{4}}}{\sqrt{15{\cdot}6}}$$

or in round numbers

$$\theta_f = \frac{(dN)^{\frac{3}{4}}}{4} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (115)$$

Expressed in words, this means that the steady or finally assumed temperature of a bearing of given diameter is greater the greater the ¾ power of the revolutions, and is greater for a given speed the larger the ¾ power of the diameter. *It is, however, quite independent of the length of the journal.*

We cannot, therefore, hope to lower the terminal temperature by lengthening the bearing. The heat generated increases as fast as the area for dissipating it does; for although by lengthening the journal the bearing pressure is diminished, yet since μ varies inversely as p, the frictional resistance and the heat generated are *pari passu* increased.

It would therefore appear that it is quite immaterial what the length of the journal is, a conclusion which seems at first sight to conflict not only with the text-books, but also with practice, and so to be paradoxical.

The explanation is as follows: Whilst it is true that the final temperature to which a bearing will rise after a long run under a given load and with a given lubricant, depends only on the diameter of the spindle and on the speed of revolution, and not at all upon the length of the journal, we have to remember that *if at that finally attained temperature and speed the pressure be too high the lubricant will be squeezed out.*

If, referring to Stribeck's and Heimann's curves—Figs. 111 and 112—we take it that the conditions attending the minimum values of the coefficient of friction are those under which the oil begins to escape from the bearing, and that for any lower speed the bearing will run dry, we may find the connection between rubbing speed and temperature on the one hand, and rubbing speed and pressure on the other, which specifies the condition of *critical* running with a complete oil film.

According to Heimann's curves—Fig. 112—from experiments in which the temperature of the bearing was varied, whilst the bearing pressure remained the same, it appears that the coefficient of friction attains a minimum for a different value of the surface speed for each different temperature, and that the value of this speed increases proportionally to the temperature, as numerically stated on the Fahrenheit scale. In fact, if v_o be the value of the speed for the minimum, and t the bearing's temperature, Fahrenheit, we find $v_o = \frac{1}{2}t$.

From Stribeck's and Heimann's curves—Fig. 111—drawn for experiments in which the bearing pressure was varied, the temperature remaining unaltered, the speed for the minimum coefficient is found again to be proportional to the pressure. For Stribeck's Sellers bearing $v_o = \frac{1}{2}p$, and for Heimann's—solid—bush bearing $v_o = \frac{1}{16}p$.

Thus v_o varies as t when p is constant, and v_o varies as p when t is constant; therefore, when both p and t vary, v_o will vary as their product; or $v_o = Apt$.

The value of A in this expression is found to be different not only as between Stribeck and Heimann, but also as between Heimann's constant temperature and his constant pressure experiments.

For Stribeck's Sellers bearing we get

$$v_o = \frac{pt}{35} \text{ (constant temperature).}$$

For Heimann's bush bearing we get

(*a*) For constant pressure experiments $v_o = \dfrac{pt}{82}$

(*b*) For constant temperature experiments $v_o = \dfrac{pt}{190 \text{ to } 240}$.

Now, Sommerfeld has deduced mathematically an expression for this very speed for the minimum coefficient of friction, in terms of the bearing pressure, p; the radius of the journal, r; the difference between the radii of bearing and journal, δ; and the coefficient of viscosity, k.

It is of the form

$$v_o = \mathrm{K}\frac{\delta^2}{r^2}\frac{p}{k} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (116)$$

We also found above that the coefficient of viscosity for machinery oil is, for temperatures between 68° Fah. and 160° Fah., well represented by the expression $k = \dfrac{1}{(t^\circ - 60^\circ)}$ or $k = \dfrac{1}{\theta^\circ}$ (units = pounds, feet, and deg. Fah.).

Substituting this for k in (116) we obtain

$$v_o = \mathrm{C}p\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (117)$$

where C is a constant, *varying as the square of the ratio of the original slackness of fit to the size of the journal.*

We prefer, therefore, to use the formula $v_o = \mathrm{C}p\theta$, which has a theoretically sound basis, rather than the form $v_o = Apt$, which was deduced from the experimental data of Stribeck and Heimann; more especially as the value of A differs so much for the different sets of curves. We ought, no doubt, to expect to find a difference for this constant as between the Sellers type of bearing used by Stribeck and the bush bearing of Heimann, but the large variation of A in the two sets of (pressure and temperature) curves given by Heimann for the same type of bush bearing seems to point to some discrepancy, such as a considerable difference in the original amount of slackness allowed between journal and brass in the two runs.

We shall, therefore, use for our purposes in designing bearings the expression $v_o = Cp\theta$, with $C = \frac{1}{40}$ for the value of the constant for machinery oil; and when written in the form

$$\theta_c = \frac{40v_o}{p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (118)$$

we see that *it gives the critical value* (θ_c) *of the final rise of temperature, above which, under the given conditions as to surface speed* (v_o feet per minute), *and bearing pressure* (p lb. per square inch), *the oil will be in danger of leaving the bearing and allowing it to run with metallic contact of the rubbing surfaces.*

What we now propose to do is to fix the proportions of journals by reference to the minimum points of the experimental friction curves already discussed, assuming them to indicate the critical conditions as regards speed, pressure, and temperature of the bearing, below which it is unsafe to run.

We first calculate the terminal rise of temperature of the bearing by the expression $\theta_f = \frac{(dN)^{\frac{3}{4}}}{4}$ (115). We then equate this to the value $\theta_c = \frac{80v}{p}$ of the above critical temperature (118); and so obtain a relation between the length, the load, the diameter, and the speed of revolution.

We thus have

$$\frac{(dN)^{\frac{3}{4}}}{4} = \frac{40v}{p}$$

Substituting $\frac{\pi dN}{12}$ for v, and $\frac{P}{ld}$ for p, we may write this

$$\frac{(dN)^{\frac{3}{4}}}{4} = \frac{40\pi d^2 Nl}{12P}$$

from which we obtain

$$l = \frac{P}{40d^{\frac{5}{4}}N^{\frac{1}{4}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (119)$$

In applying this method to the design of the front bearing of a lathe spindle, it may be remarked that, in the formula $\theta = \frac{(dN)^{\frac{3}{4}}}{4}$, the constant was determined by the consideration of the final temperatures assumed by Stribeck's Sellers bearing; and that the coefficient of emissivity—upon which that constant depends—will certainly be greater for the massively proportioned plummer blocks of a headstock, and the corresponding value of θ will be less.

We shall for the present neglect this and adhere to the above formula, the result being that we shall err on the safe side. The diameter of the spindle is fixed by considerations of strength, and its value as a function of the height of centres has been fully discussed above. For our present purpose we may take it as $d = \frac{h}{3}$. The spindle speed for which the bearing must run cool is clearly the highest for which the lathe is designed; and for this we have given the rule $Ng = \frac{3600}{h}$ for high-speed steel lathes.

Thus we find that the product dN *is a constant for all sizes of lathe.* In fact, $dN = 1200$, and $(dN)^{\frac{3}{4}} = 205$.

Hence $$\theta = \frac{(1200)^{\frac{3}{4}}}{4} = \frac{205}{4} = 51° \text{ Fah.}$$

So that (51° + 60°) = 111° Fah. is the terminal temperature which all front bearings should assume, no matter what may be the size of the lathe, if the above experimentally determined coefficients are correct.

If this calculated temperature ($\theta = 51°$) is higher than that at which the oil begins to squeeze out under the given conditions of speed and pressure, the bearing will run dry and seize.

Therefore $\frac{(dN)^{\frac{1}{2}}}{4}$ must be not greater than, or at most equal to, $40\frac{v}{p}$.

That is
$$\frac{40\pi d^2Nl}{12P} = 10\frac{d^2Nl}{P} = 51$$

or
$$l = \frac{51}{10}\frac{P}{d^2N} = 5\frac{P}{d^2N} \quad . \quad . \quad . \quad . \quad . \quad (120)$$

Taking $P = 40h^2$ (the cutting force due to the standard cut $a = \frac{h^2}{6400}$),

$$d = \frac{h}{3}$$

and $N = \frac{3600}{h}$, we have

$$l = 5\frac{40h^2 \times 9 \times h}{h^2 \times 3600} = \frac{h}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (121)$$

The length of the front bearing should be equal to half the height of centres.

This line has been plotted on Fig. 104, p. 184, and agrees very well with the spots from practice for high-speed steel lathes.

Now this is the "critical" length of the journal, or that for which the oil film would be just on the point of being squeezed out after several hours' running at the highest spindle speeds. If any margin of safety is to be allowed, the length should apparently be greater than half the height of centres; greater, that is, than that allowed by practice! The explanation is obvious. Lathes are hardly ever run for so long as to allow the journals to attain their maximum temperatures. Experiments have shown that an ordinary Sellers bearing requires to be run for more than three hours to reach its highest temperature; and when we have regard to the relatively greater massiveness of the proportions of shafts, housing, and caps in a lathe headstock, it will readily appear that a much longer time will be required before the final temperature is reached—that, in short, it will hardly ever be attained, except in special lathes.

The more correct method of attacking the problem is to determine the temperature reached by the bearing and journal after the greatest usual length of run at high speed under normal loading, as a function of the thermal capacity of that portion of the shaft and its housing which receives the heat, and of the rate of external heat emission due to the difference of temperature between these and the surrounding air.

The dimensions of the journal should be such that the temperature attained is always less than that at which the oil film approaches its critical thickness under the given condition of speed and bearing-pressure.

Let the bearing of diameter d and length l inches have run for t minutes at N revolutions per minute under a load P lbs.; and let the excess of its temperature over that of the surrounding air at that time be θ °Fah. In a short succeeding time dt (minutes) let its temperature rise $d\theta$ (°Fah.).

Then the quantity of heat generated by friction per unit of time (a) is equal to the sum of the quantities of heat absorbed by the material of shaft and housing (b); and dissipated by conduction and external radiation (c).

The quantity a equals the load multiplied by the coefficient of friction into the speed of rubbing and divided by Joule's equivalent. Or

$$\frac{\mu \mathrm{P}\pi d\mathrm{N}}{12\mathrm{J}}dt = \frac{1{\cdot}8\sqrt{\pi d\mathrm{N}}}{p\theta\sqrt{12}}\,\frac{\mathrm{P}\pi d\mathrm{N}}{12\mathrm{J}} = \frac{1{\cdot}8}{\mathrm{J}}\left(\frac{\pi}{12}\right)^{\frac{3}{2}}\frac{(d\mathrm{N})^{\frac{3}{2}}ld}{\theta}dt$$

$$adt = \frac{(d\mathrm{N})^{\frac{3}{2}}}{3230}\,\frac{ld}{\theta}dt \text{ (B.T.U. in } dt \text{ mins.)} \quad . \quad . \quad . \quad . \quad (1)$$

The quantity b equals the thermal capacity of so much of the volume of housing and shaft as is affected by the rise in temperature multiplied by the rate of temperature rise in °Fah. per minute. That is, if $\mathrm{C}ld^2$ be the number of cubic inches of metal which are heated (C being a constant varying from 2 to 10), γ be the average density of that metal, say 0·275 lbs. per cubic inch, and S its average specific heat, say 0·11,

then $$bd\theta = \mathrm{S}\gamma \mathrm{C}ld^2d\theta = 0{\cdot}03\ \mathrm{C}ld^2d\theta \text{ (B.T.U. in time } dt) \quad . \quad . \quad (2)$$

The quantity c really consists of three parts—(1) that which is conducted away by the material of the shaft (proportional to d^2); (2) that which is conducted away by the material of the housing (proportional to ld); and (3) that which is convected away by the circulating oil (proportional to the quantity fed in).

In the present case we shall ignore part (3). The other two are finally dissipated by external radiation from either the shaft or housing surface; and we shall take it that the heat emitted is proportional to that surface, to the difference of the temperatures of the surface and of the external air, and to the time. That is, if the external surface be Bld (where B is a constant varying from about 5 for Stribeck's Sellers bearing to about 10 for the more massive housings here in question) and l is the coefficient of *external* emissivity in B.T.U. per square inch per minute, we have—

$$\mathrm{C}dt = e\mathrm{B}ld \,.\, dt \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

With regard to the value of the emissivity coefficient, the coefficient $e = \frac{1}{220}$, on p. 199, which was determined from the experiments made by Stribeck, was that of "internal emissivity" and gave the heat dissipated per minute per square inch of *projected bearing area.* The value of e we now propose to use will therefore be the old value divided by B. As B was 5 for the Sellers bearing, our e will now equal $\frac{1}{220\mathrm{B}}$ or $e = \frac{1}{1100}$ B.T.U. per minute per square inch of external housing area, per degree Fah. difference of temperature between surface and air, is the coefficient we shall now employ.

We have then

$$\frac{(d\mathrm{N})^{\frac{3}{2}}}{3230}\,\frac{ld}{\theta}dt = 0\ 03\mathrm{C}ld^2d\theta + \frac{\mathrm{B}}{1100}ld\theta dt \quad . \quad . \quad . \quad . \quad (4)$$

This may be written

$$\frac{(d\mathrm{N})^{\frac{3}{2}}}{3230} = 0{\cdot}03\mathrm{C}d\theta\frac{d\theta}{dt} + \frac{\mathrm{B}}{1100}\theta^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Taking $d = \frac{h}{3}$; $N = \frac{3600}{h}$; $c = 10$ and $B = 10$; we get

$$\frac{(1200)^{\frac{3}{2}}}{3230} = \frac{h}{10} \cdot \theta\frac{d\theta}{dt} + \frac{1}{110}\theta^2 \quad . \; . \; . \; . \; . \; . \; . \quad (6)$$

or

$$\frac{130}{h} = \theta\frac{d\theta}{dt} + \frac{1}{11h}\theta^2 \quad . \; . \; . \; . \; . \; . \; . \; . \quad (122)$$

This is of the form—

$$y\frac{dy}{dx} + my^2 = n \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

and its solution is—

$$y^2 = 2C\epsilon^{-2mx} + \frac{n}{m} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (8)^1$$

Thus

$$\theta^2 = \frac{2C}{\epsilon^{2mt}} + \frac{n}{m} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (9)$$

When $t = 0$, the bearing has the air temperature; *i.e.* $\theta = 0$ and $2C = -\frac{n}{m}$

Therefore

$$\theta^2 = \frac{n}{m}\left[1 - \frac{1}{\epsilon^{2mt}}\right] \quad . \; . \; . \; . \; . \; . \; . \quad (10)$$

In equation (122) $n = \frac{130}{h}$ and $m = \frac{1}{11h}$

$$\therefore \; \frac{n}{m} = \frac{130 \times 11h}{h} = 1430 \quad \text{and} \quad 2mt = \frac{1}{5{\cdot}5h}$$

Application to a lathe of large size with

$$h = 100 \text{ in. and } t = 180 \text{ mins.}$$

$$\epsilon^{\frac{t}{5{\cdot}5h}} = \epsilon^{\frac{180}{550}} = \epsilon^{0{\cdot}328} = 1{\cdot}387$$

$$(\log \epsilon^{0{\cdot}328} = 0{\cdot}328 \times 0{\cdot}4343 = 0{\cdot}142 = \log 1{\cdot}387)$$

$$1 - \frac{1}{1{\cdot}387} = 1 - 0{\cdot}722 = 0{\cdot}278; \; \sqrt{0{\cdot}278} = 0{\cdot}527$$

hence

$$\theta_{180} = 38 \times 0{\cdot}527 = 20° \text{ Fah.}$$

$$t_{180} = 20 + 60 = 80° \text{ Fah.}$$

That is: a 100-in. centres lathe should rise in temperature to 80° Fah. after a three-hours' run at $N_s = \frac{3600}{h}$ revs. per minute, no matter what the load may be (provided, of course, it is not great enough to break the oil film).

---

[1] *Vide* Boole's "Differential Equations," 1865, p. 38.

The form $y\frac{dy}{dx} + my^2 = n$ can be reduced to the linear form $\frac{dy}{dx} + Py = Q$ as follows:—

Put $\quad y^2 = 2v$; then $y\frac{dy}{dx} = y\frac{dy}{dv}\frac{dv}{dx} = y\frac{1}{y}\frac{dv}{dx} = \frac{dv}{dx}$

Substituting, we get $\frac{dv}{dx} + 2mv = n$.

The solution of $\frac{dy}{dx} + Py = Q$ is

$$y = \epsilon^{-\int P dx}\left[\int \epsilon^{\int P dx} Q dx + C\right]$$

Here $P = 2m$ and $Q = n$;

Hence

$$v = C\epsilon^{-2mx} + \frac{n}{2m}$$

and

$$y^2 = 2C\epsilon^{-2mx}\frac{n}{m} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

The final temperature it will attain is

$$\theta_f = 38 \text{ or } t_f = 98° \text{ Fah.}$$

This value is less than that previously obtained (*vide* p. 201) because we have assumed the external area of the housing to be *twice as great* as for the Sellers bearing used by Stribeck.

For a 10-in. lathe, with t again = 180 minutes, we shall have

$$\varepsilon^{\frac{180}{55}} = \varepsilon^{3\cdot28} = 26\ 3$$

$$(\log \varepsilon^{3\cdot28} = 3\cdot28 \times 0\cdot4343 = 1\cdot42 = \log 26\cdot30)$$

$$1 - \frac{1}{26\cdot3} = 1 - 0\cdot038 = 0\cdot962; \ \sqrt{0\cdot962} = 0\cdot98$$

Hence $\theta_{180} = 38 \times 0\cdot98 = 37\cdot3$; and $t_f = 97\cdot3$

That is: a 10-in. lathe front bearing will have attained practically the final temperature (98° Fah.) after a three-hours' run at its highest spindle speed.

Large lathes are liable to run continuously for longer periods than small ones.

We should, therefore, take a larger value of t than 180 minutes for large lathes; and it will probably be best in every case to design the bearing for the final temperature.

This will lead to a result not very materially smaller than that already obtained on p. 202 *ante;* so that the rule

$$l = \frac{h}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (121)$$

is again afforded as *giving the length required to keep the bearing pressure below the limit at which rupture of the film is likely to take place at the higher spindle speeds.*

Case II.—We have next to consider the second of the extreme conditions under which a lathe spindle has to run, as specified above—that, viz., of carrying a heavy load or sustaining a heavy cut at a very slow speed.

What now happens is the entire collapse or tearing asunder of the film of lubricant, owing to the slow speed at which the bearing is being worked. Stribeck's and Heimann's curves of speed and pressure, at constant temperature, show that, after the minimum points are passed, as the speed still further diminishes, the coefficient of friction begins to rise very rapidly, and that, whatever the load may be, all the curves converge upon the value 0·15, the coefficient of "dry" friction for greasy metals, determined by Morin.

In the case of such imperfectly lubricated journals, Beauchamp Tower found that 250 or 300 lb. per sq. in. was the highest pressure at which his bearing would run without seizing. Assuming therefore a pressure of 200 lb. per sq. in. of projected bearing area as *safe* for the front journal of a lathe when loaded with a force due to its standard cut—viz. $40h^2$ lb.,—we obtain the very simple rule—

$$l = \frac{40h^2}{200d} = \frac{40h^2}{200h/3} = \tfrac{3}{5}h \quad . \quad . \quad . \quad . \quad . \quad . \quad (121\text{A})$$

for the length of the front bearing.

It will be of interest to inquire whether it is physically possible to give such large dimensions to the front bearing of a lathe as would be necessary to prevent the oil-film from being broken at the low speeds and high pressures which frequently occur.

The Sellers bearing gave the relation $v_o = \frac{1}{2}p$, and the bush bearing $v_o = \frac{1}{16}p$ for these points. If, therefore, the surface speed of the journal be v feet per minute, the bearing pressure may not exceed $2v$ lbs. per sq. in. for a bearing in halves, and $16v$ for a solid or bush bearing, if metal-to-metal contact of journal and bearing is to be avoided.

Applying this principle to the actual case, we must find what surface speeds are met with in practice, and what bearing pressures we encounter when the lathe is running at its slowest velocity of rotation and taking its deepest cut, or carrying its heaviest load.

It may fairly be assumed that every lathe will at times be called upon to take a finishing cut at a speed of 18 ft. per minute upon a heavy job of face plate diameter of the greatest length the lathe can take in, and of which one-third the weight will be taken by the front journal. We obtain, therefore, the following value of load and speed:—

The weight of the job assumed, $2h$ in diameter and $6h$ long, is

$$\frac{\pi}{4}(2h)^2 \times 6h \times 0{\cdot}3 \text{ lb.} = 5{\cdot}66h^3 \text{ lb.}$$

One-third of this will be $2h^3$ (allowing for the weight of face-plate, etc.), or $W = 2h^3$.

If the bearing be taken of diameter $\frac{h}{3}$, the rubbing speed is $\frac{18 \times h}{3 \times 2h} = 3$ ft. per minute; and, according to the above table, the safe load is therefore from 6 lb. to 48 lb. per sq. in. of projected bearing area. But the actual bearing pressure will be, if we assume the usual value $\frac{h}{2}$ for the length of the journal,

$$p = \frac{2h^3}{\frac{h}{2} \times \frac{h}{3}} = \frac{12h^3}{h^2} = 12h; \text{ i.e. } 120 \text{ lb./sq. in.}$$

for a 10-in., and 600 lb./sq. in. for a 50-in. lathe.

We see, therefore, that it is impossible to give the large dimensions to the front bearing that would be necessary to prevent the oil film from being broken at such slow speeds; and, *as a matter of fact, lathe spindles turning at the slow speeds used for heavy cuts inevitably run metal-to-metal with their brasses, giving rise to the high frictional resistance corresponding to the coefficient of friction of* 0·15 *for greasy metals.*

The work thus spent and wasted on friction and wear may, as worked out above, amount to from 2 per cent. to 11 per cent. of the total useful work expended on cutting, at the standard rate $\left(\text{area} = \frac{h^2}{6400}\right)$. From ¼ to 9 horse-power is, therefore, wasted, according to size, on the friction *of the front journal alone* when the lathe is running at these slow rates with a heavy job between centres.

Even if the job be light, and the thrust on the front journal that due to the standard cut only, we saw above that 2½ per cent. of the useful work is spent on friction on any size of lathe when the speeds are so low as to squeeze out the oil film.

We are here face to face with a very serious loss of power, and a correspondingly large amount of wear of the spindle and brasses in the front bearing, *not at all due to high speeds of rotation of the spindle;* and it is owing to this that the elaborate arrangements for adjustment of the spindle in a lathe headstock have to be provided.

It is impossible to give enough area in the front bearing of a lathe headstock to prevent metallic contact of journal and brass at the slower speeds, if dependence is placed upon the lubricant being carried in by the ordinary action of the shaft's rotation, the supply being automatic.

By using a separately driven force pump, however, and injecting a stream of moderately heavy oil into the bearing *near the place where the pressure is greatest,* it is possible to raise the journal off the brass even when at rest, and to keep it floating with a film of oil interposed between itself and the bearing when in motion, be that motion as slow and the load as high as it may.

If metal-to-metal contact can in this way be prevented at slow, and by the ordinary methods at high speeds, there seems to be a possibility that *wear may be entirely eliminated.*

If this be so, it follows *that adjustments for wear are unnecessary,* and instead of the elaborate and expensive designs of front and back bearings which are now used we may expect that a simple solid bush of ample thickness and of a length not greater than $2\sqrt{h}$, as indicated above, will meet every requirement.

Such a solid bush, of hard bronze round the steel spindle, has a great deal to recommend it from the point of view of accuracy of fit, solidity, and stiffness, as compared with the intricate methods of adjustment which are now common.

The Length of the Back Bearing.—The diameter $d_1 = \frac{h}{4}$, and N is, as before, $\frac{3600}{h}$.

Therefore $$d_1\text{N} = 900$$
and $$(d_1\text{N})^{\frac{3}{4}} = 165$$
Therefore $$\theta^1{}_{,} = \tfrac{165}{4} = 41° \text{ Fah.}$$

(Or the terminal temperature is 101° Fah.) This must be equated to $\frac{40v}{p}$.

Therefore $$41 = \frac{40\pi d^2\text{N}l}{12\text{P}} = 10\frac{d^2\text{N}l}{\text{P}}$$

or $$l = 4\frac{\text{P}}{d^2\text{N}} \quad \ldots\ldots\ldots\ldots \quad (126)$$

Taking $\text{P} = 20h^3d = \frac{h}{4}$, and $\text{N} = \frac{3600}{h}$

$$= 4\frac{20 \times h^3 \times 16}{h^2 \times 3600} = 2 \times 0{\cdot}18h$$
$$= 0{\cdot}36h \quad \ldots\ldots\ldots\ldots \quad (127)$$

The Length of the Back Bearing should be $\frac{1}{3}$ of the Height of Centres.—This line has been plotted on Fig. 104, and may be compared with that drawn to fit the spots from practice. This has the equation $l = \frac{h}{3} + 2$ . . (128)

CHAPTER XXVI

THE MAIN SPINDLE (*continued*)—THRUST BEARINGS

Thrust Bearings.—In small lathes the methods generally adopted for taking the thrust and pull of the spindle are shown in Figs. 93 and 94. In Fig. 93 the washer and check nuts on the spindle at the right-hand side of the bearing take the thrust, whilst the pull is taken by the washer and nuts on the rotating and sliding bush at the left-hand side of the bearing, any lateral movement of this bush being prevented by the check nuts on the end of the spindle. In Fig. 94 the thrust is taken on the pin in the thrust block at the tail end of the headstock. The block is fixed to the headstock by studs or bolts, and the thrust pin is adjusted and held in position by the nuts on each side of the block. The pull of the spindle is taken by the check nuts on the end of the spindle, which bear against the bush in the cheek. To prevent the turning of the nuts by the friction between them and the bush, a washer pinned to the spindle, so that it will rotate with it, is introduced between the nuts and the bush, as in Fig. 93. Without this there is a tendency for the nuts to turn, and lock or release the spindle, depending on the direction of rotation and hand of the thread.

For convenience in construction, and to reduce the distance between the front driving wheel on the spindle and the face plate, and the overhang of the latter, the thrust bearing is generally placed at the tail of the headstock. The pull and thrust should be taken as near to one another as possible, so that, in the event of the spindle or bearing heating and lengthening, jamming and excessive friction are reduced to a minimum. With this object in view, Messrs. Wm. Sellers and Co., Philadelpha, U.S.A., devised an arrangement similar to that shown in Fig. 96. Here the thrust and pull are taken on each side of a hardened steel collar pinned to the spindle. The thrust block takes the form of a cup, and is bolted direct to the facing on the tail cheek of the headstock. The thrust pin here takes the form of a ring of hardened steel, and through it the spindle may pass in order to receive the feed wheel or the driving wheel to the reversing gears, which may be placed outside the cheek without undue overhang. The cup is screwed to receive the thrust ring, which is provided with a locking nut, and bears against the collar on the spindle. The spindle is slightly reduced in diameter to form a shoulder against the collar, and so transmit the thrust. On the end of the bush in the headstock cheek is pinned a hardened steel washer, against which the collar on the spindle bears to take the pull.

A similar arrangement to the above is shown in Fig. 97, where multiple steel and phosphor bronze washers are used to reduce the liability to seize.

In this case the cup and thrust block is combined with the nut on the end of the adjustable bush, and a hardened and ground steel washer is pinned to the inside of the cup—which would be difficult to grind or harden—to receive the thrust of the spindle. A similar washer pinned on the outside of the cup, and bearing against another on the spindle, supported by the cheek nuts, takes the pull of the spindle.

The thrust bearings shown in Figs. 99 and 100 are almost identical with that of Fig. 94, with the exception that provision is made at two places in each case to take the pull on the spindle, viz. on the left-hand side of the front and back cheeks. In Fig. 100 the nut on the right-hand side of the rear bearing is intended to assist the pin to take the thrust. A recessed phosphor bronze washer is introduced between the tail of the spindle and the thrust pin, which can be readily replaced. A split nut provided with a lock bolt takes the place of the usual pair of nuts. In large sizes the keys for these nuts are clumsy to handle, making a delicate adjustment very difficult, as both keys have to be worked together when locking. With the split nut adjustment can be more easily made, and the nut locked by an ordinary spanner. Both these methods have been largely used by several firms in heavy lathes; but in view of what has already been said about the expansion of the bearings and the spindle, the design of Fig. 100 is not good practice in this respect. Where this design has given no trouble, the rubbing speed has been low, and there has been abundance of bearing surface, and consequently little or no heating. As a general rule, however, all the nuts have to be eased off when the lathe gets cold, and they only bear when warmed up by running. The mechanic fails to take into consideration this change in the condition of the parts, and concludes that it is next to impossible to adjust such bearings. In some cases it has been only when the troublesome thrust bearing has been withdrawn that the machine has worked satisfactorily. The design is altogether unsatisfactory, and is the outcome of a desire on the part of the makers to err on the safe side by securing in a cheap way an abundance of bearing surface. If provision is made by copious lubrication all the necessary surface can be easily obtained by one surface. This is equally true both for the thrust and the pull, although the former is generally greatly in excess of the latter. Some makers prefer, for heavy lathes, the design shown in Fig. 101, which is really the marine propeller shaft thrust block. The thrust is here distributed over several collars, and the pull is taken by the washer and nuts on the tail of the spindle.

Whilst conical roller bearings have been used to some extent by several makers to take the thrust of the spindle, their adoption has been anything but general, owing to the objectionable end thrust on the rollers. These bearings are designed with the apices of the cones meeting at the centre of the shaft; and to keep the end thrust of the rollers within reasonable limits, so that the outer ends may not be injured, the angle of the cone does not exceed 6½ deg. Fig. 113 illustrates a bearings of this type, by the Standard Machinery Company, Providence, R.I., and the annexed table gives particulars of their recommended sizes, loads, and speeds. These suggest that the working load is inversely proportional to the speed of revolution, and that for any one speed the load increases almost in proportion to the square of the roller diameter. Experiments to determine the coefficient of friction for these bearings have been few; but the average value appears to be somewhere about 0·0025.

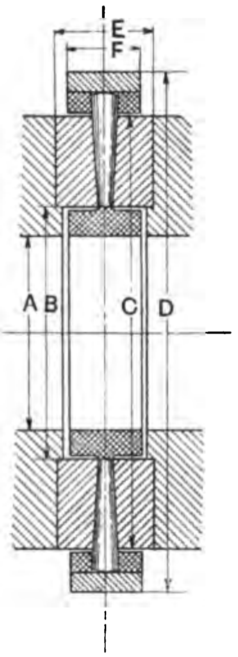


FIG. 113.

Ball Thrust Bearings.—Ball bearings are now largely used to take the thrust of the spindle in high-speed and carbon steel lathes, and seem to give every satisfaction, even in comparatively big machines. The repeated failures of this type of bearing in its early days were no doubt due to irregularity of size or shape of the balls and plates alike, to indifferent construction and to lack of uniformity in the composition of the materials employed.

Some of the leading ball bearing manufacturers now guarantee the balls to be within one ten-thousandth of an inch both as to absolute diameter and as to sphericity. Great care must be exercised in the construction and fixing of these bearings, as in such hardened steel parts the fit must be absolute. They are satisfactory only so long as the balls and races are in perfect condition. There is little or no appreciable wear in these bearings. After running for years their destruction is brought about more by the breaking up of the surfaces than by wear. The conclusions of a long series of experiments and of experience by the Hoffmann Manufacturing Company have led them to recommend the working loads for various speeds as shown on Fig. 114. The

Particulars of Roller Thrust Bearings by the Standard Machinery Co., R.I.

Diameter of shaft.						No. of rolls.	Angle of rolls.	Safe pressure on bearing.		
A	B	C	D	E	F			Area of plate P.	Speed.	
									50 revs.	100 revs.
							deg.			
1 -1¼	1⅝	2⅝	3¼	¾	½	28	6½	3·338	4,200	4,200
1 9/16-1¾	2 3/16	3 9/16	4 5/16	⅞	⅝	30	6½	6·21	11,130	8,350
2 1/16-2¼	2 13/16	4 9/16	5 9/16	15/16	11/16	30	6	10·137	19,400	10,000
2 9/16-2¾	3½	5⅝	6⅝	1⅛	⅞	30	6	15·23	33,400	16,760
3 1/16-3¼	4 1/16	6 9/16	8	1¼	1	30	6	20·862	51,200	25,600
3 9/16-3¾	4⅝	7⅝	8 15/16	1⅝	1⅜	30	6	28·86	69,000	34,500
4 1/16-4¼	5¼	8½	10 5/16	1¾	1½	30	6	35·097	80,000	40,000
4 9/16-4¾	5¾	9¼	11 5/16	1⅞	1⅝	30	6	41·234	100,000	50,000
5 1/16-5¼	6⅜	10 7/16	12⅜	2 1/16	1¾	30	6	54·2	118,500	59,250
5 9/16-5¾	7	11 5/16	13 7/16	2 5/16	2	30	6	62·52	149,000	74,500
6 1/16-6½	7¾	12⅝	14⅞	2 7/16	2⅛	30	6	78·017	189,000	94,500
6 9/16-7	8¼	13½	15¾	2 9/16	2¼	30	6	89·684	224,000	112,000
7 1/16-7½	9	14⅝	17	2¾	2 7/16	32	6	104·373	256,000	128,000
7 9/16-8	9½	15½	17⅞	2⅞	2 9/16	32	6	117·808	316,000	158,000
8 1/16-8½	10	16⅜	18¾	3	2⅝	32	6	132·96	366,000	183,000
8 9/16-9	10½	17¼	19⅝	3⅛	2¾	32	6	147·12	420,000	210,000
9 1/16-9½	11	18⅛	20½	3¼	2⅞	32	6	162·98	480,000	240,000
9 9/16-10	11½	19	21½	3⅜	3	32	6	179·6	540,000	280,000

working load in pounds is plotted on a base of revolutions per minute, and indicated by the small open spots with a thick full line through their mean values. Each curve co-ordinates the permissible loads with the speed of revolution for one particular diameter of shaft. The number and diameter of balls are dependent only upon the diameter of the shaft. The values allowed are plotted on a base of shaft diameter, the number being indicated by the black spots surrounded by circles, and the diameter by the black spots only.

Plotted on the same figure are the results of experiments made by Mr. T. Hill with ball and roller thrust bearings at speeds between 1110 and 1150 revolutions per minute. The loads are plotted on a base of watts lost in friction for the various values. The ball bearing in each case consisted of 17 balls ½ in. diameter, and similarly in the case of roller bearing. To determine the size and number of balls in a thrust bearing for a given size of lathe, we proceed as follows: Taking the thrust on the spindle at 25 per cent. of the vertical force acting on the tool when taking the standard cut, we have $T = 25 \frac{h^2}{6400}$ tons, or $T = 10h^2$ lb. approximately—the same at all speeds. The highest spindle speed at which the standard cut may be expected to be taken will be that corresponding to a cutting speed $V = \frac{1}{a} + 15$ on a

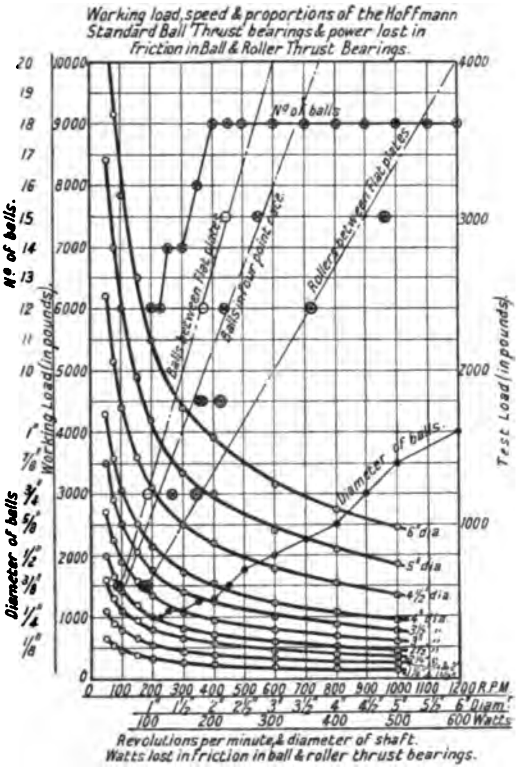


FIG. 114.

diameter not less than $\frac{h}{6}$. This gives a spindle speed equal to $\mathrm{N} = \frac{153{,}600}{h^3} + \frac{360}{h}$, *i.e.* twelve times N_t; and this is the greatest speed at which the bearing is relied on to carry the above thrust.

Using now Diagram 114, we find on the left-hand scale of loads the value of $\mathrm{T} = 10h^2$ for the thrust, and on the horizontal scale of speeds the value of N for the revolutions. Where these values intersect, we choose the nearest curve of the size of shaft, and from the horizontal scale of shaft diameters we find the number and diameter of the balls.

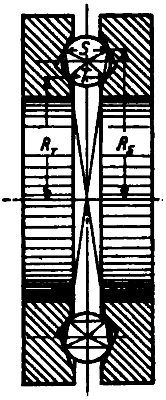


FIG. 115.

The law of variation of safe load on a ball bearing of the type shown in Fig. 115, and for which the diagram (Fig. 114) gives us the values as curves, is—

$$\mathrm{T} = \frac{4300n\delta^2}{\sqrt{\mathrm{N}}} \text{ lb.} \quad . \quad . \quad . \quad . \quad (129)$$

where n = number of balls,
δ = their diameter in inches,
N = working speed in revolutions per minute.

Now the longitudinal thrust corresponding to the standard cut is (see above) $\mathrm{T} = 10h^2$ lb. Therefore—

$$10h^2 = \frac{4300n\delta^2}{\sqrt{\mathrm{N}}}$$

or

$$n\delta^2 = \frac{\sqrt{\mathrm{N}}}{430}h^2$$

Again, we may for simplicity take, instead of $\mathrm{N} = \frac{153{,}600}{h^3} + \frac{360}{h}$,

$$\mathrm{N} = \frac{20{,}000}{h^2}$$

Substituting this, we obtain—

$$n\delta^2 = \frac{142}{430h}h^2 = \frac{h}{3}$$

Hence

$$\delta^2 = \frac{h}{3n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (130)$$

Thus, for an 18-in. lathe, if we assume eighteen balls—

$$\delta^2 = \frac{18}{3 \times 18} = \frac{1}{3}$$

$$\delta^2 = 0{\cdot}58 = \tfrac{9}{16} \text{ in.}$$

This gives a simple method of determining the size of the ball thrust bearing required for a given lathe; alternative to that by the use of Diagram 114, described above.

In Table A the results of this method are given for lathes from 6 in. to 72 in. centres.

Fig. 95 shows the type of ball thrust bearing most commonly used for lathes, although it is preferable to have the two races as near together as

TABLE A

Ball Thrust Bearings for Lathe Spindles.

Height of centres . . . . . =	6	9	12	15	18	21	24	30	36	42	48	54	60	72
Revolutions per minute $\frac{20,000}{h^2}$ =	555	247	139	89	61·5	45·5	34·8	22·2	15·4	11·35	8·7	6·86	5·57	3·85
$\frac{153,600}{h^3}+\frac{360}{h}$ . . . . =	—	250	120	70	46·5	38·7	26·2	17·7	13	10·7	8·9	7·64	6·7	5·4
Load T in lb. = $10h^2$ . . . . =	360	810	1440	2250	3250	4400	5750	9000	13,000	17,600	23,000	29,000	36,000	51,800
Diameter of back bearing = $\frac{h}{4\cdot6}$.	1·3	1·95	2·61	3·26	3·91	4·57	5·22	6·53	7·83	8·92	10·43	11·72	13·1	15·65
,, ,, ,, = $\frac{h}{4}$ . .	$1\frac{1}{2}$	$2\frac{1}{4}$	3	$3\frac{3}{4}$	$4\frac{1}{2}$	$5\frac{1}{4}$	6	$7\frac{1}{2}$	9	$10\frac{1}{2}$	12	$13\frac{1}{2}$	15	18
Number of balls	14	18	18	18	18	18	18	18	18	18	18	18	18	18
Diameter of balls	0·378 $\frac{3}{8}$	0·408 $\frac{7}{16}$	0·473 $\frac{1}{2}$	0·527 $\frac{9}{16}$	0·577 $\frac{9}{16}$	0·625 $\frac{5}{8}$	0·666 $\frac{11}{16}$	0·745 $\frac{3}{4}$	0·816 $\frac{13}{16}$	0·88 $\frac{7}{8}$	0·943 $\frac{15}{16}$	1 1	1·05 $1\frac{1}{16}$	1·15 $1\frac{1}{8}$

possible, as in Fig. 98, to reduce excess of pressure produced by heating. It is clear that, to ensure proper running of the balls in the race with a minimum of grinding action, the lines drawn through the points of contact must cross at the centre of the shaft, and that $\frac{S}{T} = \frac{Rs}{Rt}$, as shown in Fig. 115. Many are of the opinion that in ball and roller bearings there is a perfect rolling motion, and that any lubricant is unnecessary. It has been clearly demonstrated, however, by Professor Osborne Reynolds that the material gathers in front and behind the roller or ball, and thus causes relative sliding with accompanying friction. Fig. 116 shows an old type of thrust bearing with double ball races; whilst Fig. 117 gives the double ball thrust bearing of the Auto Machinery Company. In Fig. 116 the load is distributed over two sets of balls, whilst in Fig. 117 each set carries the whole load. The speed of rolling is reduced in both sets of balls in the latter, whilst in the former it is reduced to one set only.

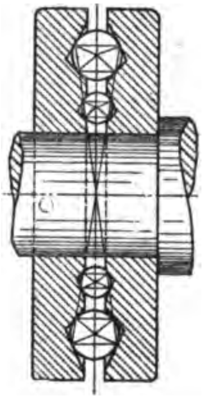


FIG. 116.

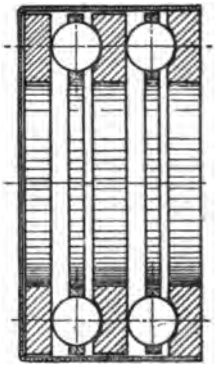


FIG. 117.

CHAPTER XXVII

BEARINGS FOR SIDE SHAFTS, COUNTERSHAFTS, AND LINE SHAFTS

Bearings for Side Shafts.—In heavy lathes the side shafts are frequently made to run in bearings similar to that of the main spindle. Generally, however, for convenience in machining, the steps are of the type shown at D and E in Fig. 118, with the split at right angles to the pull of the strap, and pegs on one or each half to prevent the bush from turning. The design shown at F is after a continental pattern, where adjustment is made by screws in the cheeks thrusting forward the pads. In small lathes the bush is usually a complete ring driven into the cheek of the headstock. It is practically impossible, even in well-proportioned bearings in ordinary use, to confine the friction to the shearing of the oil films which surround the shaft, as the supply of lubricant is more or less intermittent and the film discontinuous; consequently, there is considerable work lost by the shaft sliding in the bearing. To reduce this lost work roller bearings have been introduced where the sliding friction of the loaded surfaces has been converted into a rolling motion. The rollers are frequently simply built around the shaft and carried by a casing. In this case there is a sliding motion between the rollers themselves, and as it is essential for the satisfactory working of the bearing that the rollers shall always remain parallel to the shaft, they must be parallel in themselves, otherwise they will slew under the load and produce a grinding action. The lack of truth of the surfaces and uniformity of size of the rollers has generally been the cause of failure of this type of bearing. From these considerations it is apparent that the metals employed must be of uniform character and hardness. Very good results have been obtained where the rollers have been made of a medium carbon tool steel hardened to an ordinary spring temper, and the cases of a similar quality of steel, but tempered as hard as possible. Where cast iron has been used for the latter the rollers simply ground it into powder, and soft steel is soon laminated and compressed, thus enlarging the bearing. For high speeds balls are to be preferred to rollers, which are inclined to be noisy.

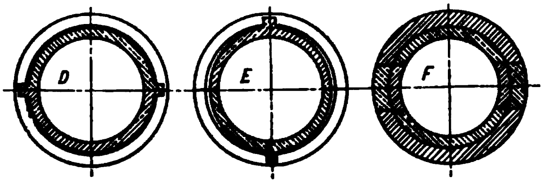


FIG. 118.

Professor Stribeck, in his investigation of ball and roller bearings, concluded that in the former the equivalent projected area was equal to one-fifth the number of balls, multiplied by the square of the ball diameter, whilst for roller bearings the equivalent projected area was equal to one-fifth the number of rollers multiplied by the length and diameter of the same.

The average of a large number of experiments made on the power consumed to drive the shafting alone in a well-appointed machine shop in which ordinary lubricated bearings were in use gave as a result that about 12½ per cent. of the total power developed was required, and a similar amount for the countershafts. Where roller bearings have been substituted this loss has frequently been reduced 25 per cent.

The conclusions arrived at by Professor Goodman in his investigation of the starting and running friction of roller bearings are as follows:—

Load.—That the friction per ton decreased as the load increased. Under the pressure of 1000 lb. at 40 revolutions, the bearing friction was at the rate of 8·512 lb. per ton of load; but under a pressure of 10,000 lb., at the same speed the friction had decreased to 3·584 lb. per ton of load.

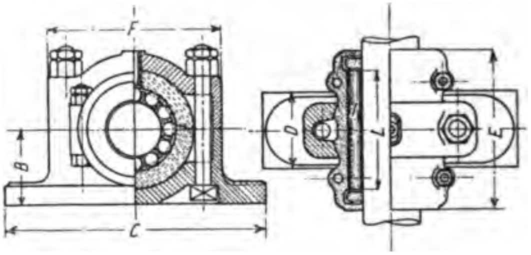


Fig. 119.

Speed.—That speed has very little effect upon the coefficient of friction. In the case of a load of 1000 lb. the coefficient was constant between the speeds of 40 and 480 revolutions per minute; and in the case of a load of 10,000 lb. it varied from 0·0016 at 40 revolutions, to 0·0013 at 160 revolutions, and remained constant at this figure to 480 revolutions, the highest speed attained in the trials.

Starting Effort.—That the slight friction to be overcome in starting was practically the same as in running.

End Thrust.—That the end thrust in a correctly constructed bearing was "practically nil."

The Empire roller bearing shown in Fig. 119 is fitted with a gun-metal cage, consisting of two end rings connected by parallel bars, within which the solid rollers are so confined that they remain parallel to the shaft, and thus reduce the tendency to cross-wind. For convenience in erecting the cage is made in halves, one part being securely dovetailed into the other. The rollers are of a high-quality steel, and of about three shaft diameters in length, and are carried by a circular steel casing, having a universal swivel seat, which fits the outer cast-iron pedestal. Here the principal sliding motion is between the rollers and the cage, which rotate with the shaft, although the cage takes none of the load. The leading dimensions of this bearing are shown in Fig. 120. This design of bearing is found to give much greater satisfaction than that with the free rollers. Some makers prefer to fit a hardened steel muff on the shaft, to prevent lamination of the same by the hardened rollers. If the rollers are soft they lengthen, and are a constant source of trouble.

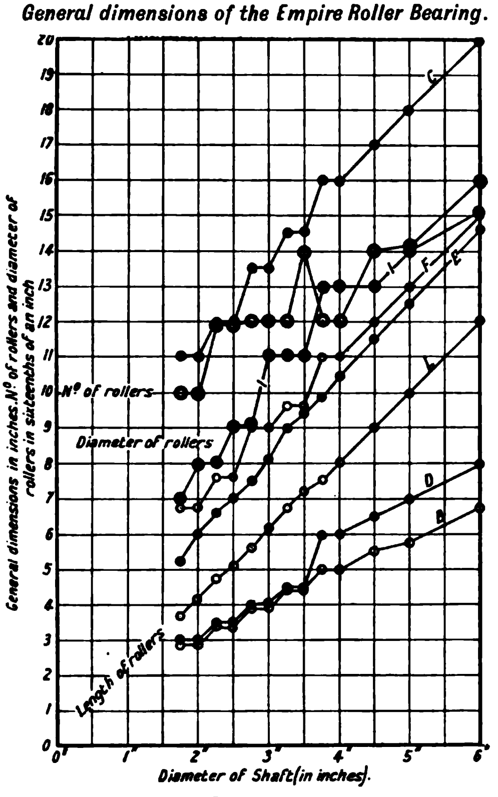


FIG. 120.

In the Kynoch bearing, shown in Fig. 121, the casing is of cast iron and in halves, accurately fitted to prevent relative movement. Instead of the solid continuous roller there is a series of short independent rollers, made of a special quality of sheet steel, and formed by rolling the blank into a hollow

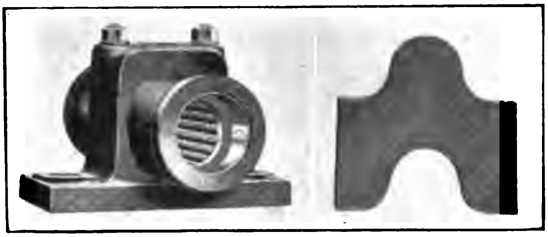


FIG. 121.

cylinder. The advantages claimed for this construction of roller are that the fibre of the steel is always in the direction of rotation, and this gives them an amount of elasticity which is never overcome by the load they have to sustain; that they are flexible. The rollers not being "hardened" do not laminate the shaft. To reduce the amount of friction between the rollers and the guiding cage there are only three compartments in the same.

Bearing Proportions for Back Shafts in Headstock.—In considering this question we have to bear in mind two radically distinct conditions under which the journals may work in the bearings. First, the journals may have always an abundant supply of oil which is carried in to such an extent as invariably to preserve a complete film all around them, even at the lowest speeds at which the back shafts have to run. This carrying in may be effected either by the shaft itself dipping in a bath of oil, or by its receiving oil carried up from the bath by one or more rotating collars or loose rings; or, on the other hand, by the pumping in of oil at the place of greatest pressure in the bearing by mechanical means.

Secondly, the supply of oil may be liable to *fail* either because it is not regularly supplied, or because, even when the bearing is flooded, the speed of the shaft is too low to carry in the requisite amount of oil against the given bearing-pressure at the working temperature. In that case the oil flows round in streaks or patches, leaving portions of the journal surface altogether unlubricated or merely unctuous. We have, in fact, wholly or partially, a metallic contact of journal and brass taking place.

We may distinguish the two conditions by the names *lubricated bearings* and *greasy bearings* for the first and second cases respectively.

The second condition is the more usual in the journals of lathe back-shafts, and the one which must be provided for in the proportioning of their bearing surfaces.

Greasy Journals.—Experimental data giving results applicable to the design of such imperfectly lubricated, *i.e.* merely greasy bearings, in which the journal revolves in actual contact with the brass, are almost entirely lacking. Beauchamp Tower found that, with imperfect lubrication, the greatest "bearing pressure" $\left(p = \frac{P}{ld}\right)$ at which his bearing ran without seizing was from 258 lb. to 328 lb. per square inch. Assuming, therefore, a pressure of 200 lb. per square inch of projected bearing area as *safe* for the journals of lathe back shafts which are liable to be sometimes imperfectly lubricated, and estimating the load as a function of their given diameter d, we are led to the simplest possible formula for the length of bearing required. The force P transmitted to or from the given shaft at the radius R of the belt pulley or spur-wheel transmitting the load should be considered as *the load on either bearing;* since it is liable to become so if the belt or spur wheel is shifted close up to it. Now $PR = \frac{d^3 f}{5{\cdot}1}$; and, taking $f = 6000$ lb. per square inch, and $R = 3d$ as average values: we have

$$P = \frac{d^3 \times 6000}{3d \times 5{\cdot}1} = 400d^2, \text{ nearly}$$

Now $$p = \frac{P}{dl} = 200$$

Therefore $$200dl = 400dl^2$$

or $$l = 2d \quad \ldots\ldots\ldots\ldots \quad (131)$$

The fact that this proportion agrees with practical experience merely shows that the above assumed values for PR, f and p for such shafts have been selected of at least the right order of magnitude.

Individual instances for which P and R are actually known may be treated, as they arise, by the same method.

Lubricated Journals.—For ring oiled bearings or other cases in which an abundant supply of lubricant can always be maintained between journal and brass we proceed in the manner already indicated for the main spindle.

The critical temperature of the bearing expressed as an excess above 60° Fah. is: $\theta_c = \frac{40v_o}{p}$; v_o being the speed (ft. per min.) at which metallic contact is in danger of occurring under the given bearing pressure p (lb. per sq. in.).

The finally attained temperature of the bearing of diameter d (inches) [after running, *i.e.*, for several hours] at the speed of revolution N (revs. per min.) is (also in excess of 60° Fah.)

$$\theta = \frac{(dN)^{\frac{3}{4}}}{4}$$

For cool running, without wear, we must therefore have θ less than, or at most equal to, θ_c, *i.e.*

$$\frac{(dN)^{\frac{3}{4}}}{4} \leqq \frac{40v_o}{P}$$

Now if l is the (unknown) length of the bearing and P its given load, we have $p = \frac{P}{ld}$. Also, if the final temperature attained is allowed to be just over the critical temperature corresponding to the given speed and load, then $v_o = \frac{\pi dN}{12}$. So that we have—

$$\frac{(dN)^{\frac{3}{4}}}{4} = \frac{40\pi d^2 Nl}{12P}$$

or

$$l = \frac{P}{40N^{\frac{1}{4}}d^{\frac{5}{4}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (132)$$

The values of $p = \frac{P}{ld} = 40\ (d\ N)^{\frac{1}{4}}$ have been calculated for the various values of N and d, which usually occur in lathe back-shafts on the fast headstock; and have been plotted in the diagram annexed (Fig. 122).

Thus, *e.g.*, if we have a shaft of 2-in. diameter running at 100 revolutions per minute, the bearing pressure which must not be exceeded if a complete film is always to be maintained, is by the diagram 150 lb. per square inch. Hence if the load (P) were 1200 lb., the length of the journal would be

$$\left(l = \frac{P}{150d} = \frac{1200}{150d} = \frac{8}{d}\right) = 4 \text{ in.}$$

If the load were 2400 lb., the journal would have to be 8 in. long, and so on.

With the same diameter of shaft, if the speed were only 25 revolutions, the bearing pressure allowed would be reduced to about 100 lb.; and the length would be

$$l = \frac{P}{100d} = \frac{1200}{100d} = \frac{12}{d} = 6 \text{ in. for a 1200 lb. load}$$

If, on the other hand, the speed were 500 revs. per min., the bearing pressure might be as high as 225 lb., and the length need only be

$$l = \frac{1200}{225d} = 2\tfrac{3}{4} \text{ in.}$$

In other words, the allowable bearing pressure $p = \frac{P}{ld}$ varies between 120 lb. and 225 lb. per square inch, according to the expression, for the different speeds and diameters above indicated.

The satisfactory agreement of these bearing pressures with the figures given by Unwin, "Machine Design," vol. i. p. 248, as having been found to agree with practice, for those cases in which the load is steady in amount and direction and the rotation not reversed, as is the case with lathe back shaft journals, shows that the new theory of journal proportioning herein proposed is at least worthy of further trial.

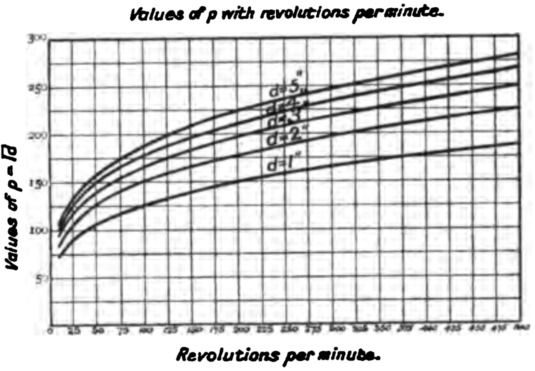


FIG. 122.

It will be noticed that, contrary to the views usually maintained, the length must be greater the *slower* the speed. This is obviously true; for the slower the speed the lower must be the bearing pressure—*i.e.* the greater l for a given d—in order to enable the journal to drag in the quantity of oil required to maintain the film at its thinnest part.

If the load on the journal may be assumed, as before, to be $P = 400d^2$, then—

$$l = \frac{400d^2}{(120 \text{ to } 225)d} = 3{\cdot}3d \text{ to } 1{\cdot}8d \quad . \quad . \quad . \quad . \quad (133)$$

The length of a well-lubricated bearing for a back shaft may be 1·8 diameters long if the speed is high, but must be 3⅓ times the diameter if the speed is so low as 80 revolutions per minute.

Bearings for Countershafts and Line Shafts.—Fig. 123 shows the standard pattern of swivel bearing of the Unbreakable Pulley Company, Limited, Manchester. These bearings are all made of a length equal to four shaft diameters, and are provided with a central ball seat which fits the sockets in the ends of the vertical adjusting screws. The nuts for these screws are cast with the hangers, and the arrangement permits of rapid and easy erection. Fig. 124 shows a self-oiling swivel-bearing by the same firm, which is interchangeable with the bearings mentioned above, so that they can be used in any of their standard fittings. In experiments made by them they found that the abundant lubrication secured with the two rings enabled them to reduce the length of the bearing from four to three diameters. It will be observed that the bearing contains the features of a well-designed self-oiling bearing—large oil reservoir with ample settlement space; no loose lids or covers; a reversible elbow which serves as a filling plug, level gauge, and emptying cock, the length of the elbow being such that the bearing cannot be overfilled; outer casing long enough to include collars which help with the lubrication; ample oil-lifting by means of a ring at either side of the centre, as shown; rings in halves, and leakage prevented by drain channel in joint. Fig. 125 shows a design of self-oiling pedestal with white or Babbitt metal lining.

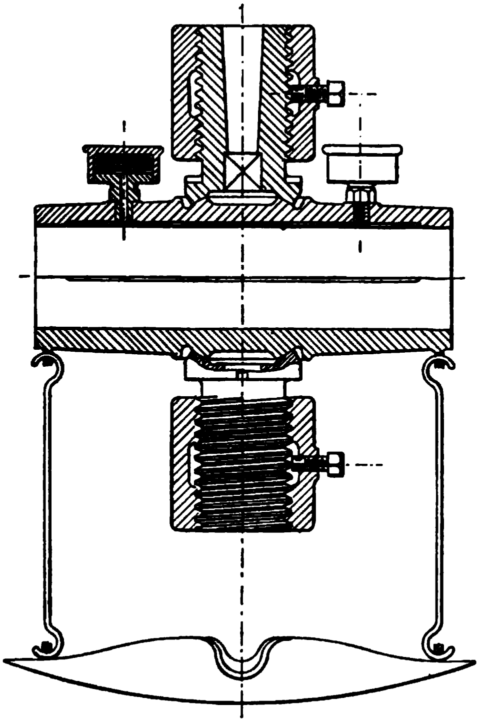


FIG. 123.

The proportions given to such well-lubricated bearings are easily justified, and shown to have a rational foundation by the use of our formulæ deduced from Stribeck and Heimann's experiments.

As already explained in the section on back-shaft bearings, we must have for the maintenance of a complete film

$$l = \frac{\mathrm{P}}{40}\mathrm{N}^{\frac{1}{4}}d^{\frac{5}{4}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (132)$$

or,

$$p = \frac{\mathrm{P}}{ld} = 40(d\mathrm{N})^{\frac{1}{4}}$$

The values of this expression have been plotted in diagram 122, and, taking as an average value suitable for the ordinary sizes and speeds of line (and counter) shafts from that figure, the value $p = 180$ lb. per sq. in., we have

$$\frac{P}{ld} = 180, \text{ or } P = 180ld$$

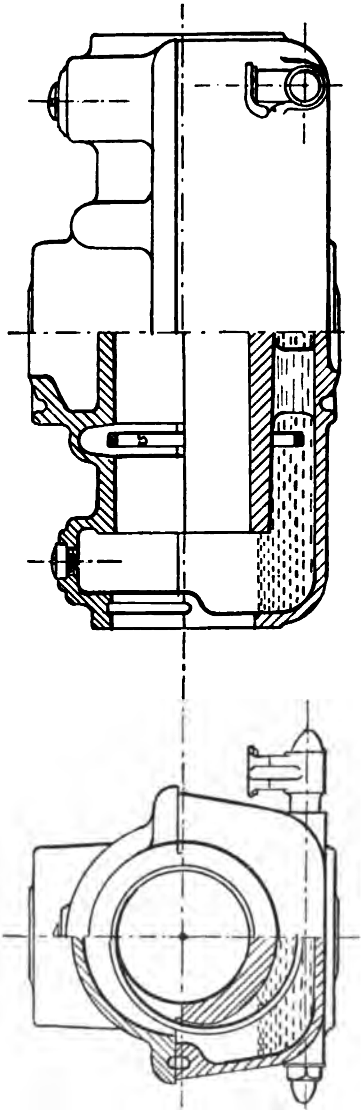


FIG. 124.

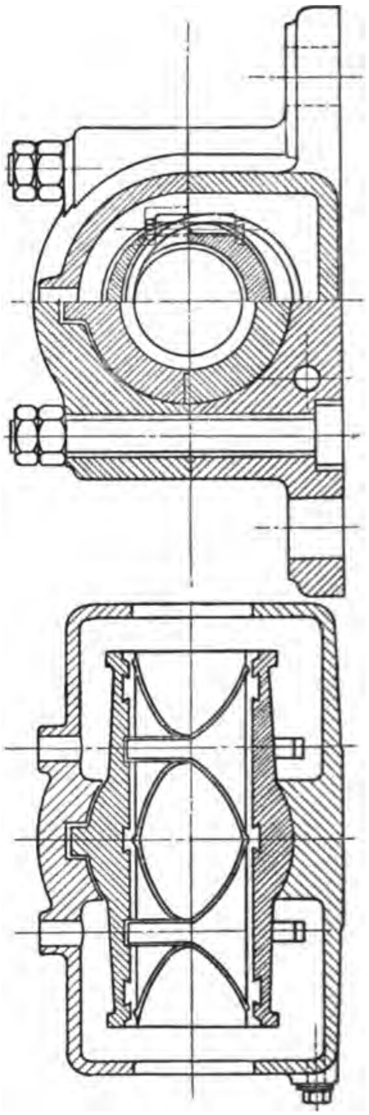


FIG. 125.

Now, in the above-mentioned system of bearings the length is taken equal to three times the diameter, which gives

$$P = 180 \times 3d^2 = 540d^2$$

This means that if those bearings are correctly proportioned the load on them must be proportional to the square of the shaft diameter.

The allowable load on a line shaft may be fixed either by a consideration of its strength or its stiffness. If we elect to work by the latter we have

$\delta = \frac{PL^3}{nEI}$ for the central deflection, where P is the sum of all the belt pulls and other loads in a length L between bearings, I is the moment of inertia of the shaft section, E is Young's modulus, and n is a constant; but the central deflection which is allowed in practice is proportional to the span (L), say $= \frac{L}{400}$

therefore
$$\frac{L}{400} = \frac{PL^3}{nEI}$$

from which we easily deduce that
$$P = \frac{nE}{k} \frac{d^4}{L^2} \qquad (134)$$

The span between bearings is in practice, however, usually proportional to the square of the shaft diameter; therefore P varies as d^2; *just as it should do if the practice of making the length of the bearing a fixed ratio to the diameter is to be justified.*

And if the span between bearings varies as the square of the shaft diameter, which in practice is nearly realized, then

$$P = cd^2$$

So that the above required condition is fulfilled; and the method of sizing the bearings for shafting, by which their lengths are made a constant number of times their diameter, is shown to be justified by experimental results rationally interpreted.

Forced Lubrication.—By the use of a pump to force the oil drawn from a reservoir into the bearing *near the point of maximum pressure* the length of the bearing can be very much diminished, *even for the slowest speeds*, especially for journals whose load and rotation directions do not change.

The rules for design above indicated no longer hold good. The finally attained temperature of the bearing will, perhaps, not be much affected by the mechanical feeding in of the oil near the place of highest pressure; but there is no longer any question of a critical velocity depending on that final temperature and the given bearing pressure. For, even with the heaviest loads and slowest speeds, the pump will always avail to preserve a film of oil between shaft and bearing at the point of nearest approach. The length of the bearing is now governed only by the necessity of avoiding the escape of the oil from the ends of the bearing and forcing it to pass round and through the place of maximum pressure.

In all probability the length need not be more than equal to the diameter; and, if this be so, it means that the use of such bearings implies the possibility of reducing the friction of line shafts *by about two-thirds of its present amount.*

The annexed figures (Fig. 126) show such an arrangement, where, however —probably for sentimental reasons—the bearing is two diameters long. It is sold by the Forced Lubrication Co., Limited, 21, John Street, Adelphi, London, W.C.

On the shaft is placed an excentric collar C. D and D^1 are end chambers to contain the oil; E is a passage to connect the end chambers together;

F is a pump plunger or thimble, made from steel tubing forced on to a solid end; G are inlet holes in the plunger for oil to pass to the inside of the pump. H is a spring to keep the plunger against the excentric; I is a non-return ball valve; and J an outlet from the pump to the shaft. L is a screwed plug to drain off spent oil and dirt, and M are leather washers to prevent the oil travelling along the shaft; N is a screwed plug for drilling and cleaning the passage E. The pump plunger or thimble F is reciprocated by the rotation of the excentric. When the plunger F is at the top of its stroke, the oil in the end chamber D escapes through the inlet holes G inside the pump. As the plunger descends the inlet holes G are cut off by the

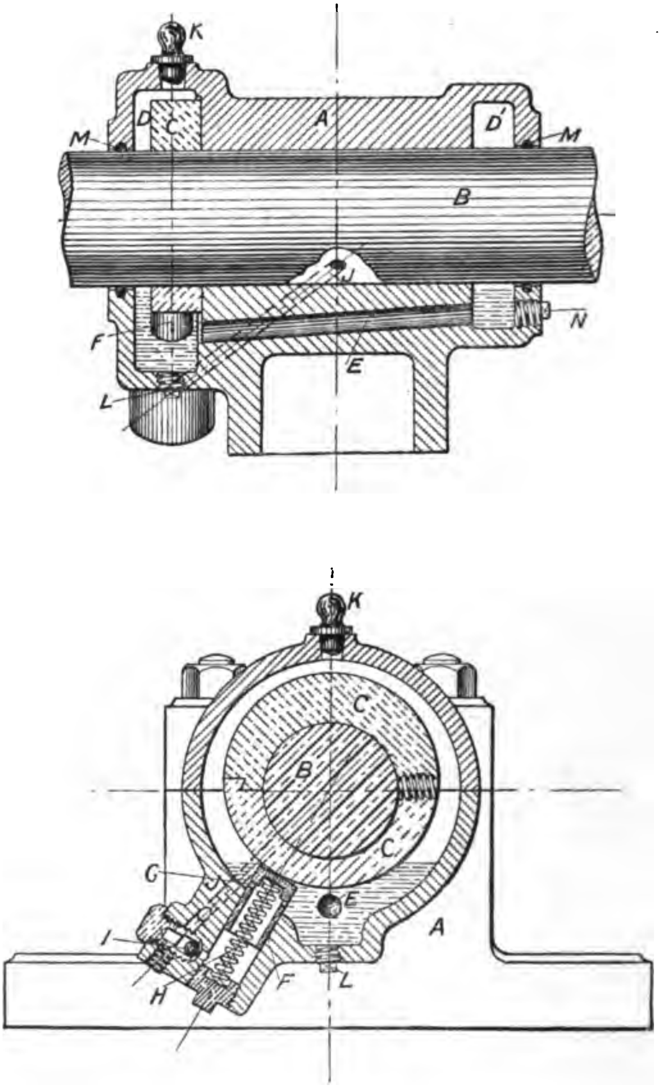


Fig. 126.

casting, and thus the oil is forced past the non-return valve I, and through the outlet J to the shaft.

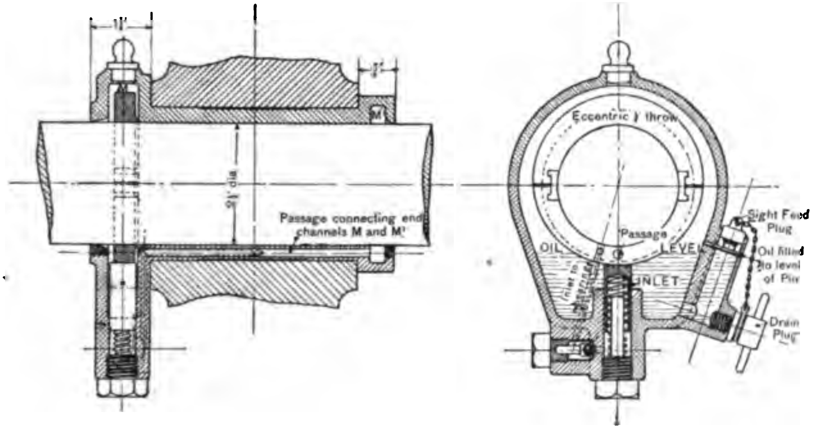


FIG. 127.

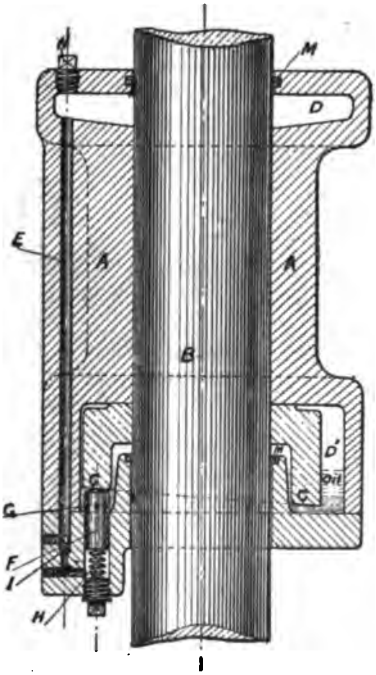


FIG. 128.—TILSTON'S PATENT AUTOMATIC FORCED LUBRICATION SYSTEM FOR VERTICAL BEARINGS.

Figs. 127 show the application of the method to an existing bearing, and Fig. 128 to a vertical shaft. With such bearings there ought to be no wear at all, except that due to the absence of an oil film at starting up.

CHAPTER XXVIII

REVERSING MECHANISM TO FEED MOTIONS

GENERALLY, where parts of machines have to be stripped to effect a change, a considerable amount of time is lost in performing the operation. If the parts be small they are liable to become misplaced or lost, and are almost sure to be damaged. Recognizing this, the leading tool makers have for some time striven to make their machines as self-contained as possible in the more essential parts. Amongst those details which have received considerable attention are the reversing and feeding mechanisms. The old method of obtaining a reverse direction of motion when screw-cutting or feeding by the introduction of an intermediate wheel in the change-wheel train has now been displaced in small lathes by a permanent reversing arrangement, generally attached to the tail cheek of the fast headstock. Such a device is now looked upon as an essential part of the machine. In the larger machines having more than one saddle, a reversing mechanism is usually attached to each, so that the direction of motion of any one saddle may be independent of that of the others. In some of the small machines, where the reversing mechanism is placed at the tail of the fast headstock, the direction of the feed is controlled from the saddle. The arrangement most commonly used, however, in small lathes is similar to that shown in Figs. 129 and 140, which is practically identical with that patented by the late Mr. Wm. Muir. The figure shows a design for an 8½-in.

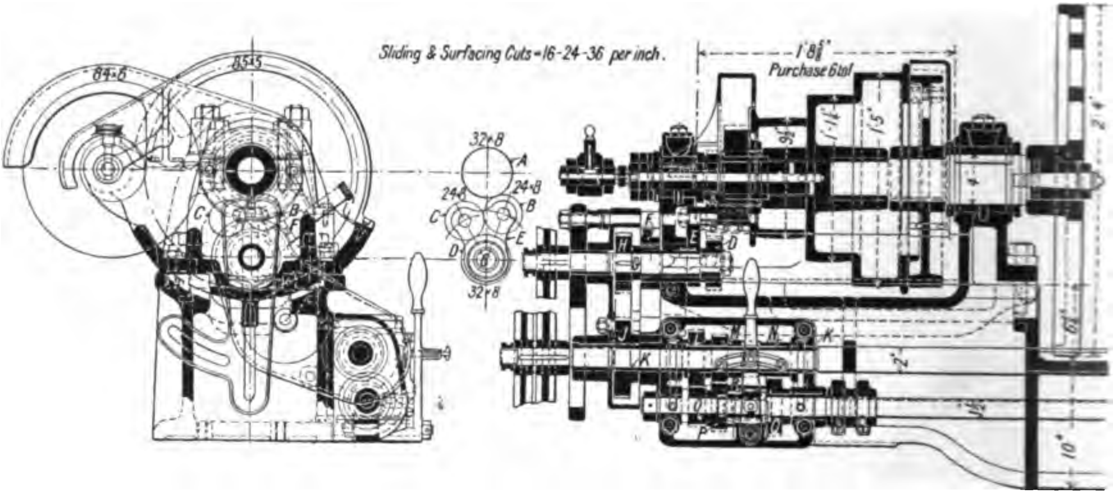


FIG. 129.—GREENWOOD AND BATLEY HEADSTOCK.

centre lathe, made by Greenwood & Batley, Limited, Leeds. The frame E has a bearing in the tail cheek, and swings about the axis of the shaft G. The intermediate wheels B and C rotate freely on the studs carried by the frame E, and either can be engaged with the wheel A on the spindle by the lever F, which is outside the cheek and keyed to the boss of E. The wheel D is keyed to the change-wheel shaft G, and engages the intermediate wheel C only. Thus the shaft rotates in the same direction as the spindle when the train A, C, D is engaged, and in the opposite direction when driving through A, B, C, D. When the lever F—and with it E—is in mid-position the gear is entirely disengaged. Provision is made in the wheel guard to receive the spring plunger pin in F, and thus locks the gear in any one of its three positions. Since the reversing wheels are almost continually in gear, and more subject to wear than any other train in the lathe, they are frequently made of steel, and machine cut. The wheels A and D are usually alike, in which case the shaft G rotates at the same rate as the spindle. Since the wheels B and C ride on the studs, and have generally short bearings, in order that the overhang of the wheels and length of the headstock may be small, they should not be less in diameter than A or D. On account of B and C acting as driver and driven wheels, and being consequently subject to twice the wear of A or D—although not on the same side of the tooth—their diameters should be twice that of these wheels. A fine pitch, and large number of teeth and diameter, reduces the irregularity of motion produced by uneven teeth, and gives a smoother motion, due to increased contact.

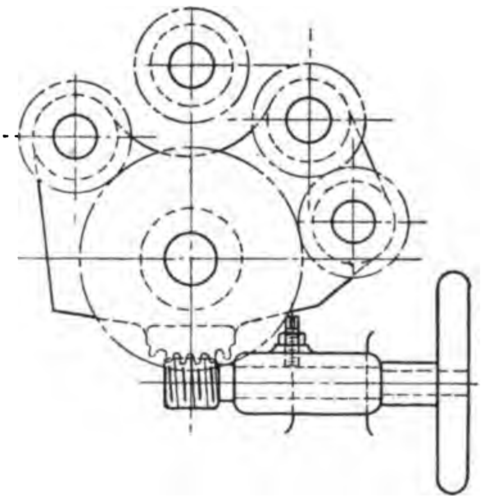


FIG. 130.—REVERSING GEAR.

In many lathes with this type of reversing gear the shaft G is provided with a bearing in the headstock cheek only. This shaft should always have two bearings, as in Figs. 129, 131, and 133, since the overhang of the wheels and drive from each end soon wears the bearing round lengthwise, and causes the shaft to rock in the same.

Some continental firms prefer the design shown in Fig. 130, where an additional intermediate wheel is introduced, thus furnishing an independent train for each direction. It is difficult to see, however, that this arrangement possesses any advantage over that just described, except that when transmitting power the wear is more uniformly distributed. By having the last wheel in the reversing train twice the diameter of the driving wheel on the spindle, the wear of the former is reduced, but the parts beyond it must be heavier to carry the increased torque than would be the case if these wheels were equal. The speed should be reduced as near to the work as possible. Again, by having a reduction of speed in the reversing gear, as shown, a wheel in the change-wheel train driving a pinion is required when cutting a thread greater than half the pitch of the leading screw, thus giving an irregular motion. These objections, together with the increased loss of power, wear in running light, and noise, more than counterbalance what advantages it may be otherwise supposed to possess.

Another form of reversing gear by the Hendey-Norton Company is shown in Fig. 131. The motion is transmitted from the spindle to the mitre wheel C, through the spur wheels A and B. The mitre wheels C and D are keyed to sleeves mounted freely on the change-wheel shaft G, and are continually in mesh with E, whose axis of rotation is at right angles to the former. The double-faced clutch F, keyed to the shaft G, and controlled by a lever at the front of the headstock, engages either C or D with the shaft G.

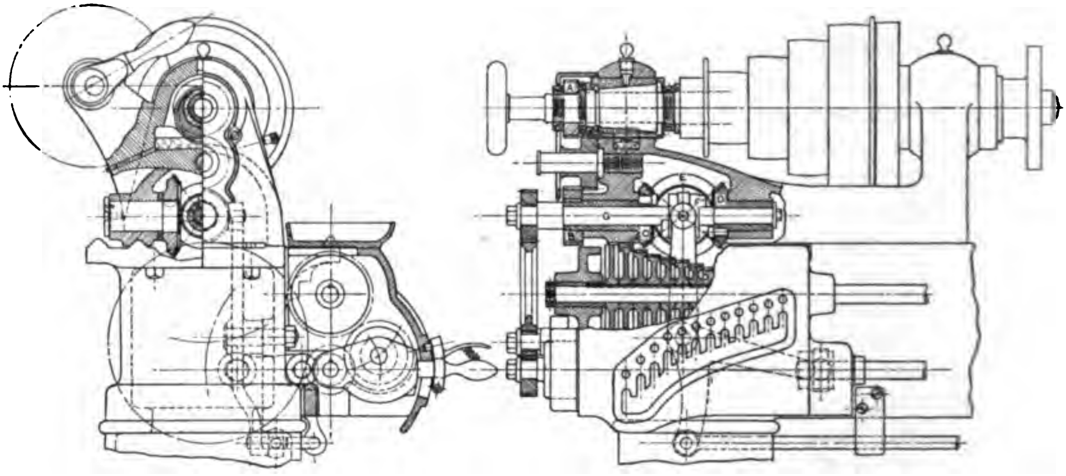


FIG. 131.—HENDEY-NORTON HEADSTOCK.

Generally the clutch has one tooth on each face considerably larger than the others, so that it can only engage the wheel at one and the same place. Thus, should the teeth of the wheels be inaccurate, the irregularities produced in the screw which is being cut are confined to the same place, and the likelihood of cross-threading, when cutting multiple-threaded screws, is reduced. The chief objection raised against this type of gear is the difficulty experienced in obtaining the same degree of accuracy in the teeth of mitre or bevel wheels as in spur wheels. Certainly this was the case when cast wheels were in vogue, and even to-day, in some of the machines in use, it is almost impossible to produce an accurate bevel wheel; yet in the best types of the more modern machines there need be absolutely no difficulty.

This arrangement lends itself also to easy and rapid control from the saddle. Some makers, however, whilst retaining this last feature, prefer to adhere to spur gearing, and adopt the scheme shown on Fig. 132. The wheels E and F are loose on the change-wheel shaft G as before, the former being usually driven direct from the spindle, and the latter through an intermediate wheel. Occasionally a supplementary shaft is employed, which is driven from the spindle, and carries the above-mentioned wheels to reduce the space occupied by them on the spindle. Although not so compact as the previous arrangement, and probably slightly more expensive at the outset, it is easier to construct, and more likely to produce an accurate screw. The design of clutch permits of an easy engagement, and tends to distribute the shock over the whole width of the teeth when entering. Its cost, however, is a frequent bar to its adoption.

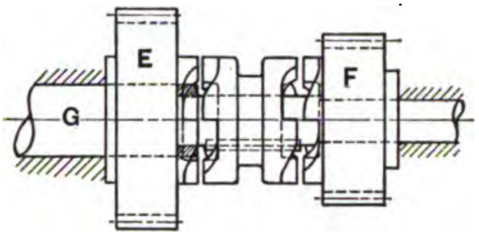


Fig. 132.—Reversing Gear.

The arrangement shown in Fig. 133 is that adopted by De Fries et Cie., Düsseldorf, to secure a series of extra coarse feeds or threads other than those obtained by the usual reversing and change gears. The wheel A is keyed to the spindle, and B to the cone sleeve, both being of the same diameter and pitch. The reversing-plate J, between the cheeks, is carried on the boss of the fixed bracket at the tail of the headstock. The intermediate wheels C and D are free to rotate and slide on the long studs carried by plate J The extra wide wheel E is keyed to the shaft G, which carries the feed cone or change wheels at its outer extremity. The arrangement as shown is similar to that illustrated in Fig. 129. The coarse feed is obtained by pushing the rod F (which is secured in the plate K and couples the intermediate wheels C and D, yet permitting them to rotate) and the wheels C and D forward, so that they may engage with the wheel B, thus increasing the speed of revolution of the shaft G in the ratio of the back gear. In the push-rod F is fitted a square-headed pin, which passes through the notch in the projection on the reverse plate carrier to prevent the wheels C and D being moved laterally when in gear with A or B. The width of the lug on the projection is equal to that of the wheels, so that there is no possibility of an accident by a partial engagement of either C or D with A and B, which

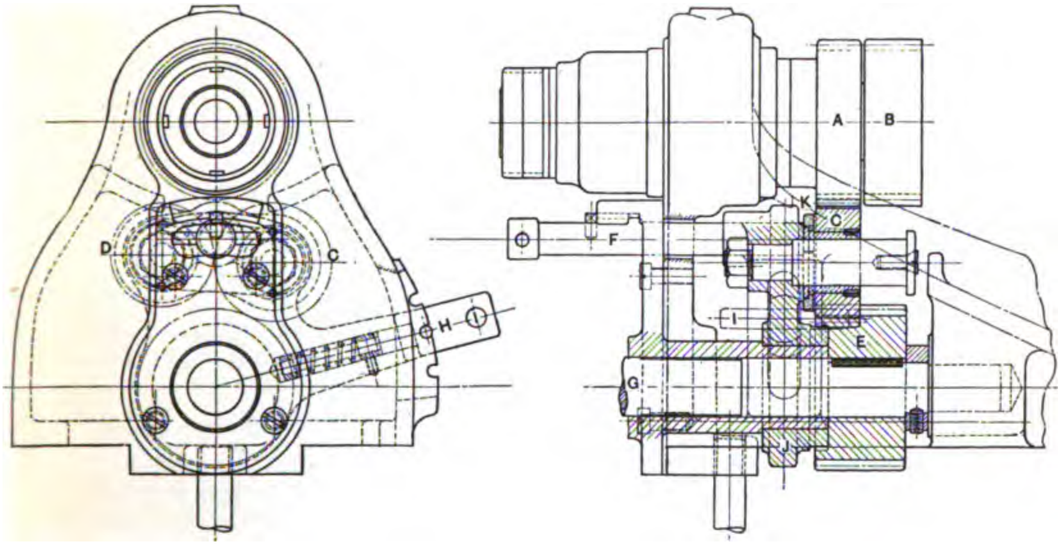


Fig. 133.—De Fries Reversing and Coarse-feed Gear.

have a different velocity ratio. The reversing plate J is held in position by the pin in the spring-plunger handle H, and the notches in the headstock cheek.

To obtain a set of coarser or finer feeds other than those obtained by the cone of feed gears, and indicated on the index plate (see Fig. 147), the Lodge and Shipley Machine Tool Company, Cincinnati, Ohio, provides a compound gear on the reversing plate, as shown in Fig. 134. The wheels H, I, K, and J form a reversing arrangement similar to Fig. 129. An additional stud carries the wheel M, which meshes with the pinion L keyed to the sleeve of the wheel K. The wheel J, keyed to the change-wheel shaft, is withdrawn, as shown, so that it gears with M for extra fine feeds, and with I for the ordinary feeds. For the extra coarse feeds the wheel K, pinion L, and wheel M are transposed, the train then becoming H, I, M, L, K, and J. It is obvious that whatever train be engaged the reversing arrangement is preserved.

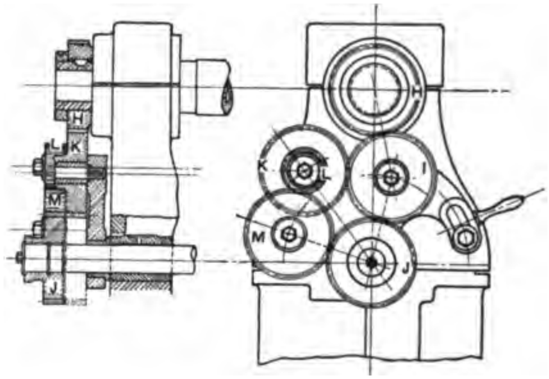


FIG. 134.—REVERSING AND COARSE-FEED GEAR.

In the larger lathes, with cone off the spindle, some British firms arrange the reversing or feed gear so that it may be driven either from the spindle or face-plate pinion shaft, thus securing a coarser set of feeds than would be obtained by the wheels when driven from the spindle.

CHAPTER XXIX

FEEDING MECHANISM TO SADDLE

(*a*) **Screw-cutting and turning Feeds both obtained by Change Wheels, etc.**—Probably no part of the lathe has received more attention within recent years than the feeding mechanism to the saddle. Not much more than ten years ago it was the general practice to obtain this motion through a cone at the rear of the fast headstock, communicating with another at the end of the feed shaft—usually placed at the back of the bed—or by means of the change wheels, which also served for screw-cutting. In the first arrangement the cone was frequently mounted on the end of the spindle or on the reverse wheel shaft, but in order to decrease the width of the feed belt the driving cone was more generally mounted on an independent stud, and run at a higher speed than the spindle by means of suitable spur wheels from the same or the reversing-wheel shaft. In this arrangement the belt acted as a safety device, as when the tool came upon an extra hard piece in the metal, or an irregular cut, the belt yielded to reduce the shock, and gradually pulled up or slipped altogether. In the second arrangement a changeable spur wheel took the place of the cone on the back shaft, and the change wheel quadrant—usually mounted on the boss of the guide-screw bearing—was designed so that the motion from the spindle or reversing wheels was communicated through the change wheels to the wheel on the back shaft. This arrangement permitted of a wide range of surfacing or sliding feeds, but for each feed it was necessary to change the wheels on the shaft, and generally those on the quadrant, causing much waste of time.

This wide range of turning feeds exceeded, however, the practical requirements; and a suitable smaller number of instantly obtained changes has been found sufficient. Where the work is small, and the number of pieces required to be either screwed or turned are few, the time taken to change the gear from feeding to screw cutting with the old arrangement would frequently amount to 20 per cent. of that required to do the work. It therefore appeared desirable that the feeding and screw-cutting gears should be capable of being instantly changed and disengaged without mutual interference; whilst, at the same time, simultaneous engagement should be impossible. The result is that the turning feed gear of to-day usually consists—in the smaller types of lathes—of a nest of wheels, placed at the fast headstock end of the machine, which permits of a change of feed by the simple movement of a lever at the front. Many makers prefer such a device to effect the sliding and surfacing feeds—turning feeds—only; others, again, arrange it so that it shall also serve for the screw-cutting feed, by reducing the number of wheel changes on the quadrant. The reversing is generally common to both feeds. In the larger lathes, however, having more than one saddle, an independent feeding mechanism is usually fitted to each saddle, thus permitting each to move in either direction independently.

Wm. Sellers & Co., Philadelphia, is one of the few American firms which has adhered to the practice of providing independent screw-cutting and turning feeds. The former is obtained in a manner after the common British practice, *i.e.* through change wheels mounted on a quadrant; but the method adopted for the turning feeds is peculiar to this firm, being through friction discs, as shown in Fig. 135. On the fast headstock spindle is keyed a large spur wheel, which meshes with a pinion on the same shaft as the friction disc A. The discs B are carried on a swinging arm, and are pulled and pushed towards one another and against the discs A and C by a strong spring. The disc C communicates with the longitudinal feed shaft through gearing, and the feed is increased or diminished by raising or lowering the arm.

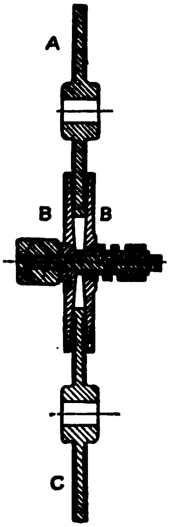


FIG. 135.—FRICTION DISCS.

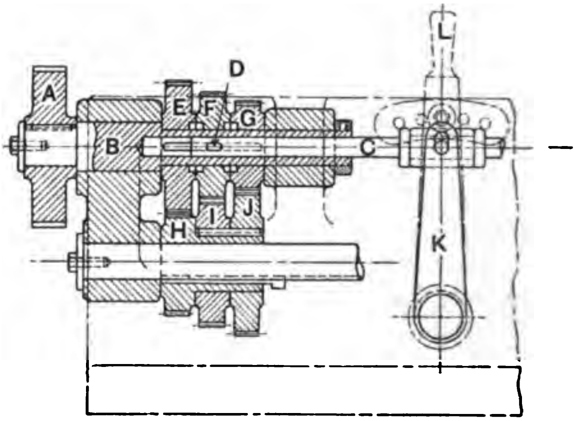


FIG. 136.

(*b*) **Screw-cutting Feed obtained by Change Wheels as before, but Feed-change Gear for the Turning Feeds.**—One of the early forms of feed-change gear arrangement still largely used is that shown in Fig. 136. The nest is placed at the back of the bed; but the change is effected by the lever L at the front. The spur wheel A, keyed to the short hollow shaft B, receives its motion from the last wheel in the reversing train. This shaft has a through elongated slot to suit the key D fitted in the push rod C; so that B, C, and D always rotate together. The wheels E, F, and G are free on the shaft B, and are provided with keyways in the body, and recesses at the ends of their hubs to receive the key. The lever L is keyed to a rocking shaft, whereon is also keyed the lever K, which actuates the rod C, and engages D with either of the above wheels. When D is withdrawn within the recesses it runs idle. The wheels H, I, and J are keyed to the feed shaft. The arrangement thus permits of three changes of feed, EH, FI, and GJ. It is obvious that by increasing the number of pairs of wheels a correspondingly increased number of changes of feed is obtained. Recesses in the bed receive the plunger-pin fitted in L, thus locking and indicating the position of the key and feed. Objections have been taken to this design on account of the wheels being continually engaged, and of their tendency to wobble or seize on the shaft, due to the short bearing surface of the loose wheels. In high-speed lathes this arrangement has been abandoned altogether by the leading firms; as the abnormal sizes of the parts—due to the weakened sections—is prohibitive.

This objection is practically eliminated in the well-known scheme where the three upper wheels are closely keyed together and to the shaft, and are yet capable of movement along the same by a conveniently placed lever. Three corresponding wheels are keyed to the feed shaft below, and spaced to allow of one pair only being engaged at any one time, or the set completely withdrawn. Thus there are no loose wheels, and only the wheels actually doing the work are in gear. In a three-change gear of this type the minimum space between the wheels on the feed shaft is equal to twice the width of the wheels plus the necessary clearance. To obtain an increased number of changes on this principle the wheels have to be unduly spread.

By compounding the gears as shown in Figs. 137, 138, and 139, Messrs. Greenwood and Batley conveniently obtain nine changes within a comparatively small space. The design is for an 8½-in. centre hollow spindle capstan lathe. The gear-box is mounted on the back of the bed, but all the changes are effected by levers placed at the front. The loose and sliding wheels have ample bearing surfaces; wheels B, C, H, I; E, F, G, and K, L, M, are formed in one piece. The wheel A on the short feed shaft is driven through suitable gearing from the spindle. The pinions B and C are keyed to the shaft, but free to slide along it by means of the lever and claw, actuated by the handle at the front of the machine. D and H, I are keyed to the free sleeve on the shaft, whilst J is free on the sleeve. Means similar to the above are employed to move H, I along the sleeve; E, F, G is free on the feed shaft, and K, L, M is keyed to it.

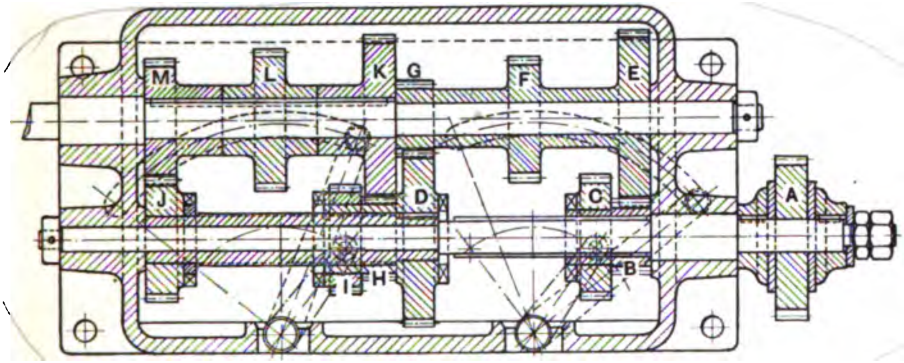


Fig. 137.

The feeds are arranged in geometrical progression, and in ascending order are as follows:—

(1) J, M. (C clutched with D, and I with J.)
(2) I, L. (C clutched with D.)
(3) H, K. (C clutched with D.)
(4) C, F, G, D, J, M. (I clutched with J.)
(5) C, F, G, D, I, L.
(6) C, F, G, D, H, K.
(7) B, E, G, D, J, M. (I clutched with J.)
(8) B, E, G, D, I, L.
(9) B, E, G, D, H, K.

Although two hand levers are employed to effect the necessary changes of feed, the arrangement is "fool-proof." No matter what position the levers may occupy, either one feed or none is obtained, as the spacing between the wheels is sufficient to allow of the drivers being entirely withdrawn without engaging with one another.

In the above designs it is inadvisable to engage the feed gear when the

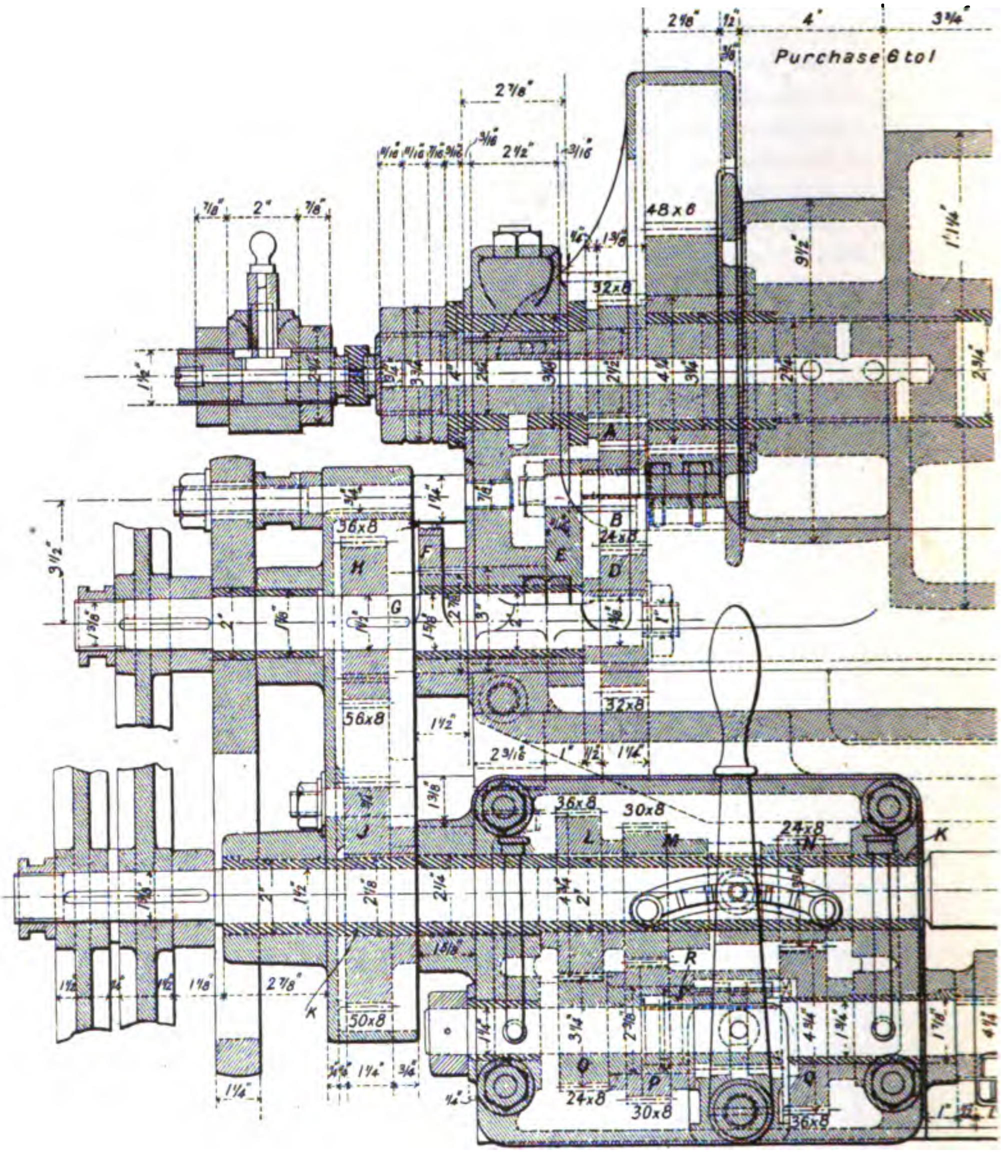


FIG. 140.—GREENWOOD AND BATLEY'S CHANGE-FEED GEAR.

machine is running at a high rate of speed, as the corners of the key, keyways or wheel teeth, which take the shock on entering, will suffer; although the operator may be cautioned against changing his feed under such conditions, yet it does not prevent him from doing so; and the mechanism being usually covered by a guard, the damage wrought each time is not apparent until it finally breaks down. Much trouble has been experienced in this way even

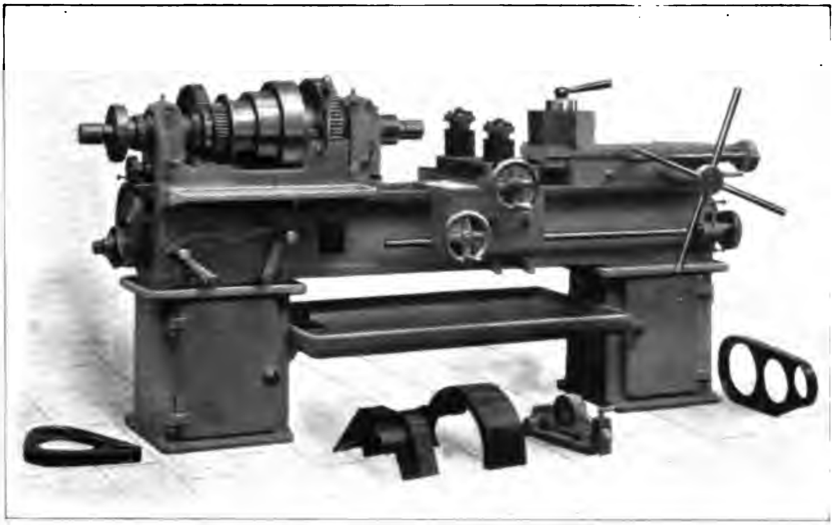


FIG. 138.

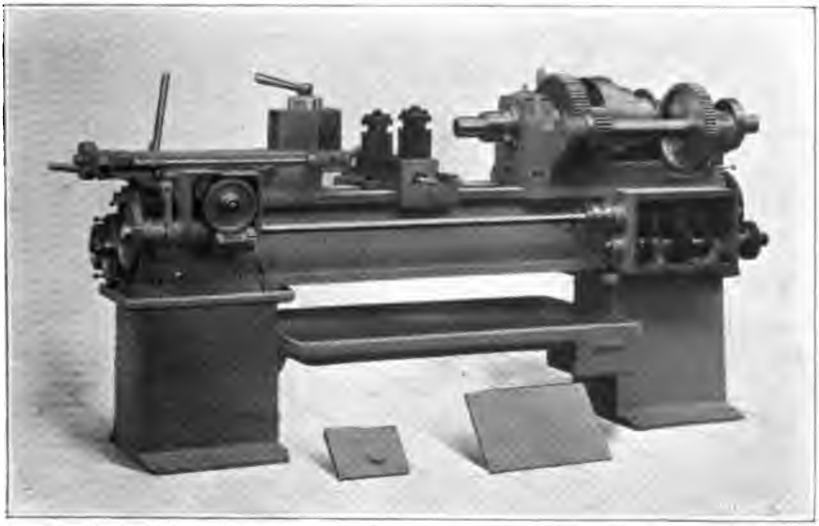


FIG. 139.—8½-INCH CENTRE LATHE (GREENWOOD AND BATLEY).

in carbon steel lathes. Under no circumstance should the feed mechanism be changed when the cut is on where disengaging gears are used.

In their 8-in. centre heavy turning and screw-cutting lathe Messrs. Greenwood and Batley have adopted the design shown in Figs. 129 and 140. The sliding wheels have been displaced by a positive clutch, and the free wheels—with large bearing surfaces—are fitted with gun-metal liners and improved means of lubrication. The feed may be changed whilst the machine is in motion without damage, as the shock at entry is distributed amongst the several teeth of the clutch, which are excessively strong in proportion to the work they have to do. When the clutch is of the saw-tooth type, the back clearance permits of an easy engagement, and the impact is more frequently taken on the full face than on the edge of the teeth. Where, however, it is desirable to feed in either direction—which is seldom, if ever, necessary in small lathes—a clutch of this type is inadmissible if the reversing mechanism is on the driving side of it. In the arrangement shown in Fig. 129, a portion of which is shown enlarged in Fig. 140, the screw-cutting is quite independent of the sliding feed. The change-wheel plate swings upon the fixed bush in the wheel guard, and about the axis of the change-wheel shaft G. The motion to the sliding feed is obtained through the wheel H, keyed to the shaft G, within the guard at the tail of the headstock, which gears through the intermediate with the wheel J on the long sleeve K. K is free to rotate on the guide screw, and has keyed to it the wheels J, L, M, and N. On the feed shaft below is keyed the double-faced clutch R, and the wheels O, P, and Q ride freely upon the same. The pinion O is made of phosphor bronze, and the wheel P rides upon its boss. A distance piece—forming part of the gear-box—keeps the loose wheels in position. It will be observed that the clutch teeth are within the boss of P, and not on the face, as in O and Q. Clearance is provided on P and R beyond the clutch teeth, to enable the latter to pass through the former, in order that it may engage with O. Clearance is also provided between the clutch teeth, so that there is no possibility of any two gears being engaged at one and the same time. This also permits of the disengagement of the feed by withdrawing the clutch from any of the wheels with which it may be engaged. The clutch is controlled by a lever at the front of the gear-box, and a spring plunger in the same fits the recesses in the face of box, and locks the gears in position. The design is for three feeds—16, 24, and 36 per inch; but it is obvious that a fourth could be obtained by two pairs of wheels on the right of the clutch instead of one.

Fig. 141 shows the development of a four-change gear device with external clutches by the American Tool Works Company, Cincinnati. They employ this device in conjunction with a cone of wheels to obtain a large variation of sliding and screw-cutting feeds. This arrangement of wheels, and the framing modified to suit the position on the machine, has been adopted by several firms as a change gear for the sliding and surfacing feeds only, as in the cases already discussed. The wheel A derives its motion from the last wheel in the reversing train, and is keyed to the shaft. The wheels C and D ride freely upon the shaft, whilst J is keyed to it, and engages C or D, which have clutch teeth on the face of their bosses. The wheels E and F are keyed to the sleeve of the pinion G, which is freely mounted upon the pin. H and

I have clutch teeth on their inner faces, and are free upon the feed shaft, but capable of being connected with it by the clutch K, which is keyed to the same. Four changes of feed are obtained by this arrangement:—

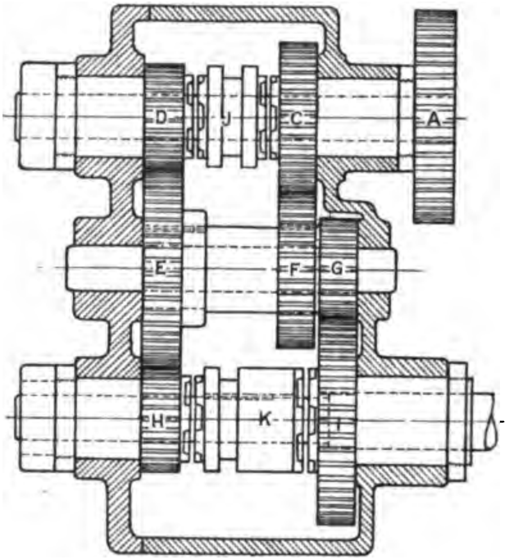


FIG. 141.

(1) C, F, E, H. (J engaged with C, and K with H.)
(2) D, E, H. (J engaged with D, and K with H.)
(3) C, F, G, I. (J engaged with C, and K with I.)
(4) D, E, G, I. (J engaged with D, and K with I.)

The clutches are controlled by handles placed outside the casing, and an index plate affixed to this shows clearly the position of the handles for each feed. Occasionally one handle only is used, and a cam on the end of the rocking shaft is made to actuate both clutches.

A very common design is shown in Fig. 142, which closely resembles that with a sliding cotter. Here, however, the key is depressed when sliding between the wheels, and its whole length is presented on entering, thus distributing the shock over a greater area, and within the same space an increased length of key and wheel-bearing surface on the shaft are obtained. The nest design shown in the figure is one situated at the back of the bed. The spur pinion A on the end of the short shaft D is driven from the reversing train through suitable gearing fixed on the headstock. The shaft D is hollow, and has a rectangular slot to receive the splined shaft B and drop key E, which swings about the fulcrum pin in B. A plunger pin, C, also carried within B, is made to thrust against the heel of the key E by a compressed spring, so that it may always enter the ways in the wheel. The initial compression of the spring can be adjusted by the screw in the end of B. The ends of the key are bevelled, and a steel washer is introduced between each

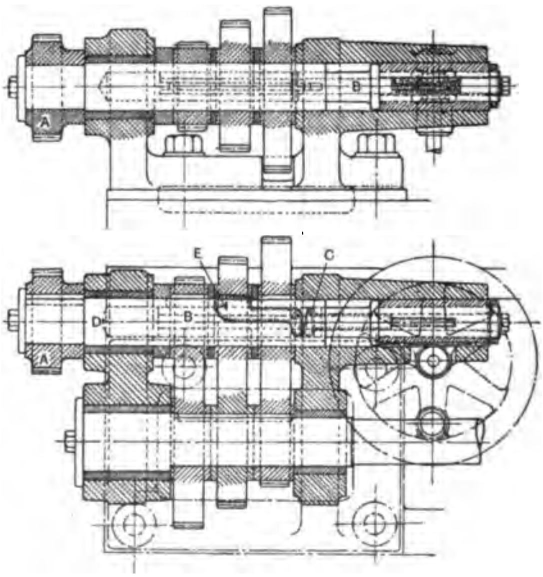


FIG. 142.

wheel, so that when the key is withdrawn or advanced it is depressed by the washers as it passes under the same, thus disengaging it from the wheels. A sliding rack, enveloping B, and carried within the bearing, is actuated by the pinion on the transverse shaft, and moves the key to the desired position. A graduated hand-wheel on the rack-pinion shaft at the front of the machine, and a plunger pin in the bed, indicates the feed, and locks the hand-wheel and mechanism in position.

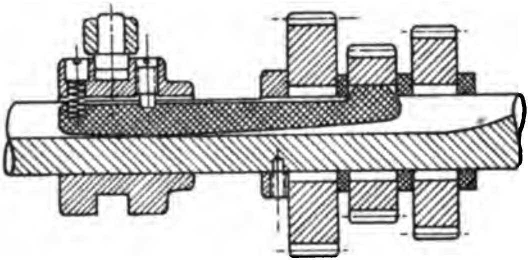


Fig. 143.

Objection may be taken to the weakened section of the driving parts, as in the sliding cotter arrangement shown in Fig. 142. The modification of these details by Alfred Herbert, Coventry, and John Lang and Sons, Johnstone, shown respectively in Figs. 143 and 144, are much to be preferred.

(*c*) **Screw-cutting Feeds obtained principally by Change Wheels, but also by the use of the Turning Feed-change Gear.**—The design shown in Fig. 144 is for an 8¼-in. centre-sliding, surfacing, and screw-cutting lathe. The wheel I is provided with a long boss, and rides freely on the guide-screw. It derives its motion from a similar wheel on the end of the spindle, and an intermediate wheel fixed at the rear of the headstock. Within the gear-box is the mitre wheel R, keyed to the boss of I, and meshing with S, whose axis of rotation is at right angles to R; S in turn gears with T, mounted freely on a long sleeve, also free on the guide-screw. A double-faced clutch, keyed to this sleeve, engages the corresponding teeth on either R or T, thus giving a reverse motion to the same. The spur wheels, J, K, L, and M, are keyed to the sleeve, and mesh respectively with Q, P, O, and N on the hollow shaft, free on the feed shaft immediately below the screw. A sliding key engages each successive pair of wheels with the hollow shaft. On the left-hand end of this hollow shaft is formed a clutch having extra long teeth, which are continually engaged with the outer set of teeth on the sliding clutch U. The clutch U has also an inner set of teeth on its right and left-hand faces which mesh respectively with the clutches fixed to the feed shaft and the change-wheel shaft W. This latter shaft is bored out at its inner end to provide a bearing for the former. When the clutch U is moved to the right, the hollow and feed shafts are made to rotate together at varying speeds, depending on the pair of wheels engaged by the sliding key. When U is moved to the left motion is transmitted through the feed wheels and hollow shaft to the change-wheel shaft W, the projecting end of which is made to receive the change wheel. This wheel in turn communicates with the wheel on the end of the leading screw through the change gears on the quadrant X—turned upwards in the figure—in the usual way. Here the screw-cutting motion is actuated through the feed gears, and four changes, having a ratio of 2, 1, ⅔, and ½ to 1, are obtained with each train of change wheels. The feeding motion is, however, independent of the screw-cutting, and the feeds—12, 24, 36, and 48 (revolutions of spindle per inch movement of tool)—are clearly indicated on the face of the gear-box for each position of the lever. With this arrangement it is impossible for the screwing and feeding motion to be engaged at the same time; and by the movement of the handle at the bottom left-hand corner the feed can be instantly changed from the one to the other whilst the machine is in motion.

The sliding key arrangement is used to a considerable extent where a large number of feeds are required; as by duplicating the sets, after the Alfred Herbert principle, and driving directly through them, or by mounting the sets abreast on independent shafts or sleeves in such a manner that they are alternately driver and driven, a very compact arrangement is obtained. In the design by Alfred Herbert, Limited, a cone of gears keyed to the driving shaft meshes with a corresponding set on a second-motion shaft, a

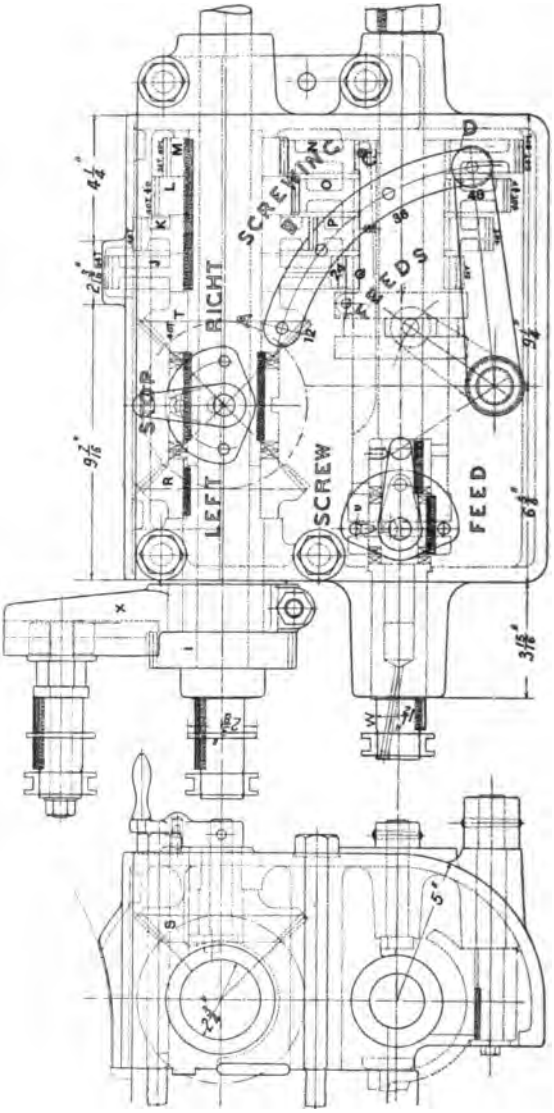


Fig. 144.—Screw-cutting Feed for 8½-in. Lathe.

sliding drop key engaging any of the set with the same. Another set of wheels keyed to this shaft engages those on the feed shaft, and is coupled to it by a second sliding key. Both these keys are controlled by a hand-wheel and cam. The number of feeds obtained for every speed to the driving shaft is equal to the number of wheels on the second-motion shaft engaged by the sliding key, multiplied by the number of similar wheels on the feed shaft.

In the design by the Carron Company, Stirlingshire, shown in Fig. 145, the variation of screw-cutting feed is effected by changing the wheel on the screw only. The wheel A is keyed to the spindle, and drives the wheel B through either of the intermediate wheels carried by the reverse wheel plate. The quadrant H swings about the axis of B, and carries the three permanent wheels, C, D, E, having sixty, thirty, and twenty teeth respectively, all of which can be made to mesh with the change-wheel F on the screw by sliding it along the same. Three threads are thus secured for every change of wheel, and with a two-threads per inch pitch guide screw, all the usual threads, from 2 to 36 per inch, can be obtained. An index and table on the face of the quadrant H give the necessary wheels for the different pitches, and their position when properly engaged. The quadrant is locked in position by a tee-headed bolt in the bed and a nut handle. To engage the turning feed, the quadrant is swung to the left, and the wheels on same made to engage the wheels on the back-shaft quadrant. Thus, by a change of wheels on the latter, together with the trains at the back of the bed, a wide range of feeds is obtained, and by means of a free wheel on the screw, both feeds are actuated by the screw without a change of gear.

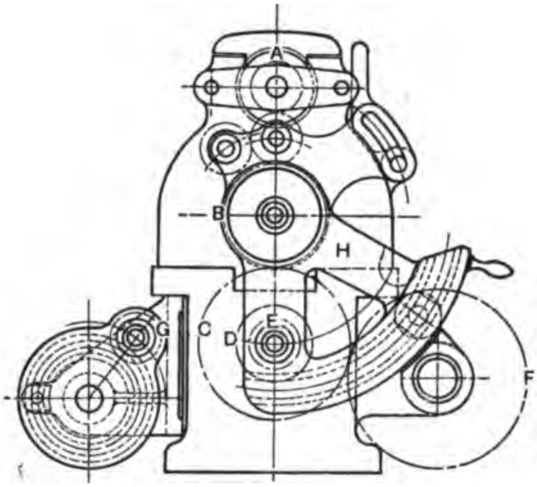


Fig. 145.—Screw-cutting Feed-change Gear.

In the arrangement by John Lang and Sons—Fig. 144—the screw-cutting feed is actuated through the fixed train of turning feed gears, and in the Carron device the turning feed is obtained through the screw-cutting gear wholly or in part.

(*d*) **Screw-cutting Feeds obtained by Feed-change Gear, together with Additional Change Wheels.**—In the majority of American lathes the turning feeds are obtained by the same mechanism as used for the screw-cutting feeds, the latter being arranged so that most of the standard threads can be cut without recourse to change wheels. Some of the leading continental firms have adopted this method in the smaller sizes of lathes, and, as in the few cases where it has been followed in this country, a modification of the Hendey-Norton device has found most general favour. The original Hendey-Norton gear is shown on Fig. 131. It consists of a cone of spur wheels keyed to the guide screw within the box at the fast headstock end of bed.

Motion is communicated to these wheels from the spindle through the change wheels on the swing frame to the pinion on the end of the sliding-pinion shaft. The latter pinion gears with the jockey wheel, which engages any of the wheels forming the cone. From this it is obvious that for every wheel in the cone a different feed is obtained, and that other sets are secured by altering the change-wheel train. A facsimile of the index plate for the 10-in. centre lathe is given herewith.

THREADS.

Stud 120	Screw 80	5	4½	4	3½	3¼	3	2¾	2½	2¼	2	1¾	1½
80	80	20	18	16	14	13	12	11	10	9	8	7	6
80	120	80	72	64	56	52	48	44	40	36	32	28	24
36	96												11½

Feeds—6 times threads per inch.

In the device by the New Haven Manufacturing Company, shown in Fig. 146, the cone is composed of bevel wheels, keyed directly to the end of the screw outside the bed. The change wheel K, keyed to the shaft P, is the driver to the system, and is driven from the spindle through a simple train of change wheels, the intermediate wheel being carried on a quadrant mounted on the boss of P for lathes below 9-in. centres. In the larger machines a compound gear is used in combination with a sliding key and wheel, whereby four changes are secured without any change of gears. The bevel wheel L keyed to P meshes with the wheel M keyed to the sleeve Q, the latter being mounted freely upon the excentric shaft. The pinion N is also keyed to Q, and is capable of sliding along the same, so that it may engage any of the wheels forming the cone. The excentric shaft is similar to that usually employed to carry the back gear wheels in the headstock, and consists of a cylindrical shaft with an excentric collar keyed to each end within the bearings. The position of the cone wheels is marked on the outside of the casing, and a hole is provided in the same to receive the lock-pin T. To change the gear, N is disengaged by the handle S and moved along Q by the claw lever until opposite the wheel given on the index plate for the desired feed, when the pin T is entered into the corresponding hole, and the wheel N again engaged. This pin, in addition to locking the gears in position, prevents the possibility of a partial engagement with any other gear. In the first arrangement with seven wheels in the cone, and four change wheels, thirty-five feeds

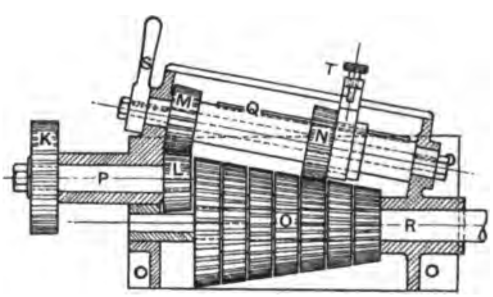


FIG. 146.—CHANGE-FEED GEAR.

are obtained; and in the second, with the compound train and eight wheels in the cone, forty threads from 1 to 30 per inch can be cut by changing one wheel only. The feeds with the latter arrangement are from 4 to 120 per inch.

(*e*) **All Screw-cutting and Turning Feeds obtained by Feed-change Gears.**—A modification of the Hendey-Norton device by the Lodge and Shipley Company, Cincinnati, is shown in Fig. 147. The last wheel of the reversing train—see Fig. 134, *ante*—is keyed to the shaft revolving between the bearings within the bed. On this is mounted the sliding wheel K, upon the boss of which is pinned the swinging arm carrying the jockey wheel L. This jockey wheel engages any one of the wheels forming the cone keyed to the shaft immediately below the above. At the end of same, outside the bed, are keyed two wheels, P and O, having a tooth ratio of 1 to 2. An arm swinging about the axis of this shaft carries the twin jockey wheels, Q, R, and S, T, each pair being fast together, but free on the spindles, and having a tooth ratio of 1 to 2. The wheels P and O mesh respectively with R and S, whilst Q, R, S, and T can be engaged with the sliding wheel M on the guide screw. Four feeds are thus obtained for every wheel on the cone, which generally contains eleven. An index plate is attached to the cover over the sliding wheel on the screw. It is marked "Thread" and "Knob" on the upper line. The sliding handle to the wheel M has a finger attached, and indicates the feed. Under "Thread" is the number the lathe will cut—2 to 32; and under "Knob" are the figures 1 to 11; thus, should the operator desire to cut any given thread, he pulls the finger to the column in which it is given, and places the knob—in the swinging arm carrying the jockey wheel to the cone—in the hole indicated. By this means forty different threads are obtained without having to ship any wheels; and with the compound gear on the reversing train—shown in Fig. 134—one hundred and twenty threads can be secured. Several will, of course, be duplicated. A copy of the index plate is given, and this may help to make the above description more easily understood.

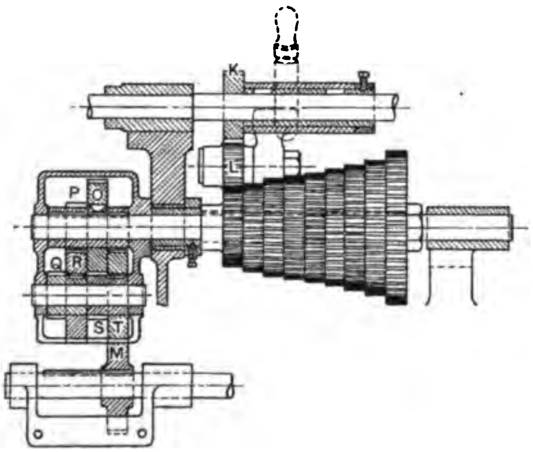


FIG. 147.—LODGE AND SHIPLEY CHANGE-FEED GEAR.

Thread.	Knob.	Thread.	Knob.	Thread.	Knob.	Thread.	Knob.
18	2	9	2	4½	2	2	1
19	3	9½	3	4¾	3	2¼	2
20	4	10	4	5	4	2½	4
22	5	11	5	5½	5	2¾	5
23	6	11½	6	5¾	6	2⅞	6
24	7	12	7	6	7	3	7
26	8	13	8	6½	8	3¼	8
28	9	14	9	7	9	3½	9
30	10	15	10	7½	10	3¾	10
32	11	16	11	8	11	4	11

FEEDS.

80 to 40	40 to 20	20 to 10	10 to 5

The turning feeds are driven through the screw, and no feed shaft is employed.

The modification due to the American Tool Company is very similar to that described above, with the exception that instead of the second series of changes, outside the bed, being derived through sliding and tumbler wheels, the clutch arrangement, previously described and shown in Fig. 141, is used, whereby four changes are obtained, and with eleven wheels in the cone, forty-four different feeds are secured. This enables all the usual threads between 2 and 32 per inch to be cut. An independent feed shaft driven through the screw is employed for the turning feed, so that there is always a fixed relation between the screw and the shaft. The turning feeds will equal, of course, the above in number, and range from $\frac{1}{8}$ in. to $\frac{1}{128}$ in. per revolution of the spindle.

In the design by the Springfield Machine Tool Company, Ohio—shown in Fig. 148—the variation of feed is effected by the change wheels on the shafts at the tail of the headstock in conjunction with the independently mounted wheels within the circular casing at the end of the bed. Motion is communicated from the spindle through the reversing wheels to the shaft K, which, like the shaft L, is provided with a clutch to engage the change wheels in the casing. The shafts extend considerably beyond the clutches, in order to receive and support the wheels. A pinion on the shaft L meshes with the wheel on the swinging arm, which in turn engages the wheel in the circular disc and on the end of the guide screw. The casing Q_1, Q_2 is in two pieces. The inner half Q_2, fixed to the boss of the guide-screw bearing, simply acts as a guard, and carries the fixed stud M, upon which the outer disc Q_1 rotates. The feed wheels have long hollow hubs, which rotate in the bearings provided in the disc, a nut on the end of the hub holding them in position, whilst the inner face of the hub takes the form of a clutch. These wheels do not mesh with each other, and although their diameters are different, their centres are equidistant from that of the stud M, to permit of any one of the series being brought opposite the guide screw by turning the disc Q_1. The end of the screw is bored out to receive the sliding shaft O, which is actuated horizontally by the rack and pinion within the bearing. A clutch P is fixed to the shaft immediately in front of the rack; and the shaft, reduced in diameter, projects beyond the clutch to provide a support for the feed wheel. When the desired gear is brought opposite the guide screw, the shaft O is advanced, and its protruding end enters the hole in the hub, prior to the engagement of the clutches on the shaft and wheel. The wheel on the swinging arm is then dropped into gear. Convex projections S, equal in height to the pitch line of the corresponding wheels, are cast on the disc, and upon these the concave projection on the swinging arm rests when the wheels are properly engaged. As the changes obtained by the wheels on the disc are of themselves insufficient, three pairs of wheels, each pair mounted in a guard as shown at R, are supplied to fit the shafts K and L. The ratios of these wheels are 1 to 1, 1 to 2, and 1 to 4. On being reversed these give five speeds, so that, with the eight wheels in the disc, forty changes are obtained, enabling all the standard threads between 2 and 56 to be cut.

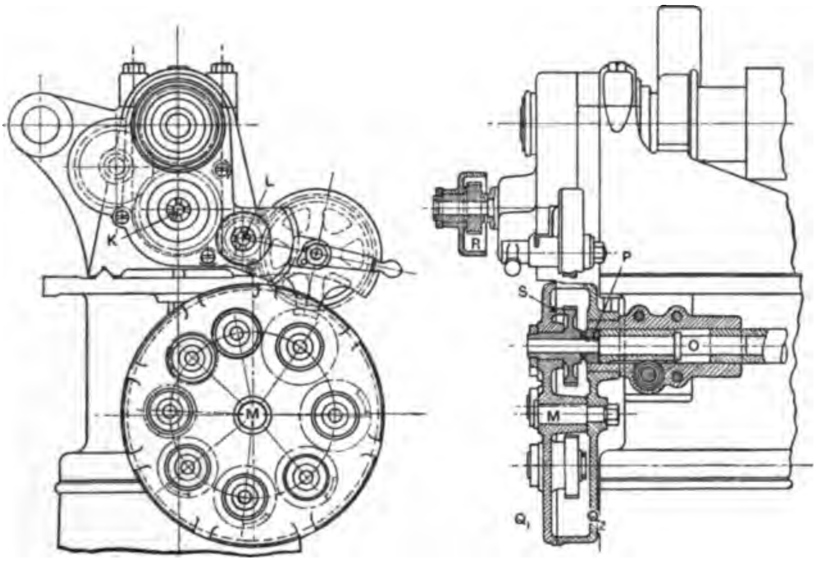


FIG. 148.—SPRINGFIELD CHANGE-FEED GEAR.

Another arrangement somewhat similar to that just described is that manufactured by the National Machine Tool Company, Cincinnati, to the design of Mr. W. L. Schellenbach, and shown in Fig. 149. Here all the usual feeds are obtained without the use of change gears. The bracket H is bolted to the tail cheek of the headstock, and the sleeve I to the bracket. Within the sleeve is the shaft J, on whose ends are keyed the wheels K and L. K is the driver to the system, and is either driven direct from the spindle, or through an intermediate wheel. The circular casing Q_1, Q_2, is in halves for convenience in construction; but is bolted together, and free to turn concentrically on the sleeve J. The boss of Q_1 is, however, excentric to J, and the two intermediate wheels, O_1 and O_2, are pinned together concentrically with the boss of Q_1, and free to rotate as one upon it. Whilst O_1 and O_2 have the same number of teeth, the pitch of the latter is slightly less than that of the former, and consequently its smaller diameter allows of larger wheels engaging it within the limits of the casing and centres of the shafts. The fact of these wheels being excentric to J permits of them being surrounded with pinions of varying diameter, the path of whose centres is concentric with J. Thus the nine pinions N_1, N_2, . . . N_9, shown in the figure, are mounted on hollow spindles within the casing Q_1, Q_2, and have their centres equidistant from that of J. The spindles S_1 are provided with clutch faces at both ends, with the exception of that for N_1. This spindle is combined with the pinion, and has but one clutch face at its outer end. All the pinions, N_1 . . . N_9, are continually in gear with the intermediate wheel O_1, O_2. Provision is made at the tail of the bracket H for the two mating wheels, P_1 and P_2, mounted on sliding clutch shafts (whose centres also lie in the same circle struck from J as that of the pinions), so that any of the spindles S_1 may be engaged with either of these shafts on the rotation of the casing. As P_1 is also always in mesh with the wheel T on the end of the guide screw, a reverse direction of feed may be obtained. The direction depends on which of the wheels P_1 or P_2 is engaged with the wheels in the casing. It is impossible for both these wheels to be engaged at one and the same time, as the knurled-headed shaft R is fitted with two rack pinions, which engage the racks on the sliding clutch shafts, and (being situated between the same) advances the one whilst withdrawing the other. The shafts extend beyond the clutch, and enter the spindle S_1 before the engagement of the clutches, thus locking the casing in position. The projecting end of the clutch shaft S_2 is bored out and split to act as a spring, and thereby hold it in position. The wheels L and M can be transposed, and the clutch shaft S_2 can engage any of the hollow spindles S_1. With L and M and clutch S_2 in position, shown in figure, motion is communicated through K, L, M, N, and O_1, O_2, to N_2 . . . N_9, any one of which can be connected with P_1 or P_2, depending on the direction of feed desired. In this way eight changes of speed are obtained. As a different speed is also secured for every hollow spindle S_1 engaged by the clutch S_2, nine further changes are obtained; and by combining both methods, 72 speeds are derived. Finally, by the transposition of L and M, 144 speeds in all may be obtained. These are, however, not all different.

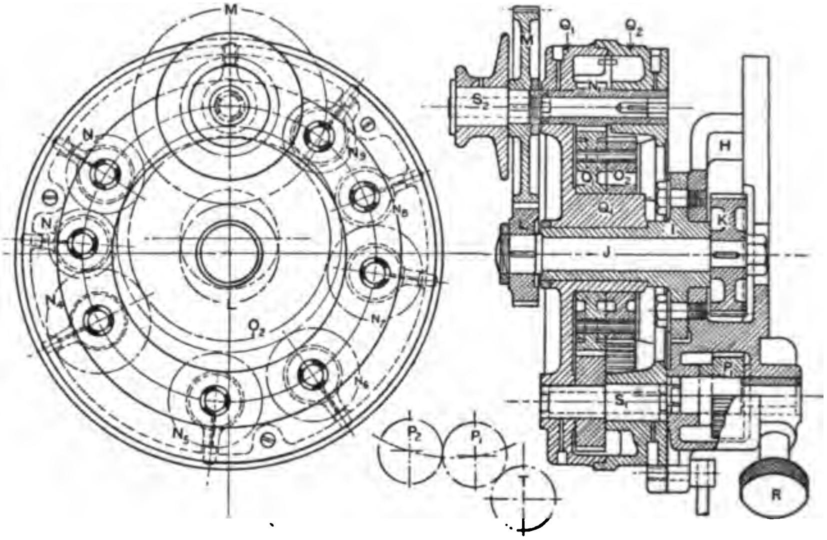


FIG 149.—NATIONAL CHANGE-FEED GEAR.

CHAPTER XXX

GENERAL REMARKS ON FEED GEAR

A CAREFUL inquiry into the number of turning feeds necessary for a general-purpose lathe, whether for high speed or carbon steel, shows that four to six are sufficient to meet all practical requirements. On the diagram given in Fig 150 we have plotted the values ordinarily supplied of the sliding and surfacing feeds for various sizes of lathes, as obtained from recent practice. Four fair curves have been drawn through the points on the assumption that

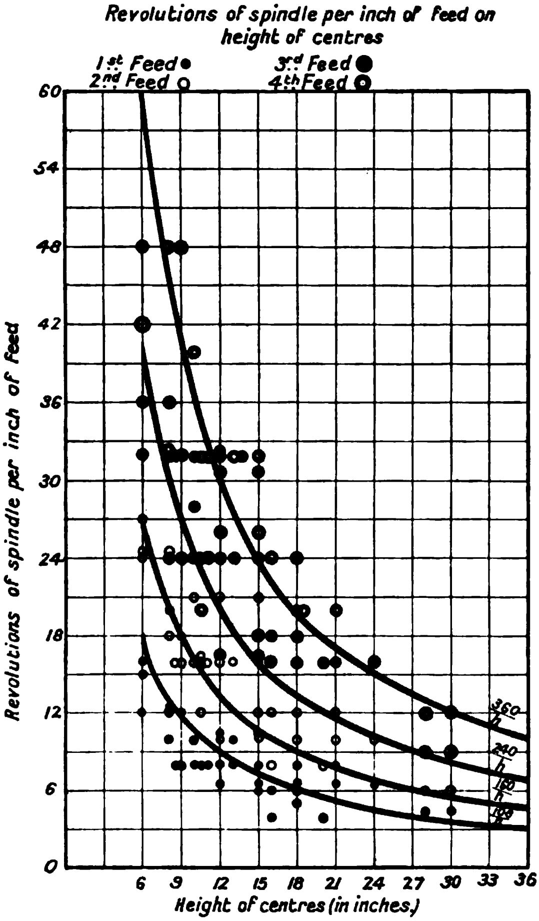


FIG. 150.

four such turning feeds are provided. These four curves have the equations $\frac{108}{h}$, $\frac{160}{h}$, $\frac{240}{h}$, and $\frac{360}{h}$ for their ordinates as a function of the height of centres, the feeds being expressed as turns of the spindle per inch of traverse advance of the tool. It will be admitted that these four curves and their corresponding rates of feed are capable of supplying all ordinary needs for general work; and it would appear, therefore, that the contention expressed above is well founded, and that the multiplication of turning feeds in excess of this, when the mechanism is independent of that for screw-cutting, is superfluous, and only results in an unnecessary addition to the cost of the machine. Where such a multiplication has been resorted to, it is usually the outcome of an attempt to follow American methods, in which a large number of turning feeds is supplied more by accident than design. In American lathes, where the feeding mechanism for screw-cutting is such that most of the standard threads can be cut without recourse to change wheels, and the turning feeds are obtained by means of the same mechanism, a large number of turning feeds are made available which never would have been provided for turning alone.

Although many recent writers have recorded their opinion that the American method is the right one to adopt, British makers generally adhere to the old plan of supplying a few turning feeds by feed-change gearing, and obtaining the screw-cutting feeds by change wheels on the quadrant, and no general attempt has been made to follow American practice. Whether this is due to the small amount of screw-cutting relatively to turning which has to be done, or to the usually adopted rule—where it is not possible to change from feeding to screw cutting after the Lang, Carron, etc., methods—of turning all the pieces completely before changing the gear to screw cutting, so that the time lost is small relatively to the increased cost of the machine, or whether it be that the present devices are unsuitable for British practice—except, perhaps, for very small machines—is somewhat difficult to say. On the other hand, the all-mechanical change-feed practice has not by any means been suddenly adopted in America, but has been gradually developed since the introduction of the Hendey-Norton device in 1892. It was, no doubt, in use to some extent prior to that; but has grown distinctly more rapidly within the last few years. Its origin may be briefly traced. The American head-stock spindle is provided with a hole, which was first intended to permit of a rod to push out the centres, which were longer and had a finer taper than is usual in Britain. It was, however, found that the hollow spindle was very useful in the turning of small work from the bar, and, as a consequence, the majority of the smaller machines are now made with draw-in collets or chucks, to allow of this class of work being rapidly handled, and the holes in the more recent machines have been increased to admit of a larger diameter of bar, equal to $\frac{h}{6}$, even in lathes up to 30-in. centres. Makers do not admit that the increase of feed changes is due to the greater capacity for bar work; yet they do not deny that the hollow spindle has been mainly responsible for the adoption of the feed-change device, and an examination of the feeds supplied to the earlier machines generally supports the idea that they were intended for small work only, and for this purpose were well suited. Thus it would appear that the chief cause for the general adoption in the United States of the Hendey-Norton system is the specialization or repetition work which has there become general and almost universal. Manufacturing on particular lines has led to close forging to size with light cuts and fine feeds easily variable, and in such cases the multiplication of feed changes which has arisen may to some extent be justifiable. Whilst the handiness of an arrangement whereby every thread upon the list can be secured by the simple movement of a lever or plug cannot be denied, it is quite certain that the interest on the first cost of a device for obtaining such a large variety of changes exceeds the value of the time saved. Nevertheless, this is the trend of present-day practice, as the more recent machines, particularly those of leading American makers, go to show. The majority of these—self-contained feed-change gear arrangements—give over forty changes, some exceed a hundred, but these need not be discussed; and although several movements are necessary to effect the change, the directions are, as a rule, so clear, and the arrangements so simple, that it is difficult for the operator to make a mistake.

No objection can be raised against them on this ground, but it must be remembered that the multiplication of joints, small wheels and shafts, is not conducive to accurate screw-cutting. If we consider the actual necessities for general work, and to what extent the present devices meet the same, we find upon careful inquiry and examination, that such screwing feeds as 22, 23, 26, 28, 32, 36, 44, 48, 52, 56, and upwards, which are invariably supplied, are scarcely used once a year in the average shop, and that most of them are altogether useless—except it be for cycle or instrument parts. This, however, is quite outside the usual run of work which comes to an engineering establishment, and consequently does not come within the scope of the lathe under consideration, and such feeds do not even appear on the American or European lists of threads. In the majority of the arrangements examined no provision is made for cutting a thread of greater pitch than two threads per inch, and in no case was it found possible to cut a square-threaded screw or worm of a fractional pitch, such as $\frac{3}{8}$ in., $\frac{5}{8}$ in., $\frac{3}{4}$ in., etc. Now, such work is not uncommon, even in America, although fractionally pitched screws are used to a less extent there than in Britain, and in the average shop this class of work will come as frequently to the lathe as a three-per-inch screw. From this it is clear that these lathes with the self-contained feed-change gear arrangements have lost their caste as general-purpose screw-cutting lathes. In some cases it would appear that practical requirements and utility have been overlooked in the search after novelty, or the desire to embody some feeds not supplied by other makers, upon which the salesman may claim attention and enlarge.

Where the change-feed gear is supplemented with a quadrant and change wheels, as in the Lang, Carron, and Hendey-Norton lathes, the general utility of the machine is not impaired, and fractionally pitched screws and such work can be done as in the ordinary screw-cutting lathe. This combined arrangement has been found to be particularly handy for small-screw work, where only a few pieces of one kind are required, and where a change of dies, etc., on the hollow-spindle lathe is not warranted. It is difficult to say what number of changes should be obtained by this means if the lathe is to be also used on general work. One thing is clear, that unless these machines are employed for a large percentage of their time on a few threads, and only incidentally on general work, there is little advantage over the old method with change wheels, as the more wheels there are in the change-gear device, the more confusing it becomes to the operator when he has to change those on the quadrant to cut, say, a fractionally pitched thread, as indicated above, or other threads not obtained by the change gear. Of course, if an index plate containing the wheels and position of lever for every possible thread is provided, this difficulty is partly removed, but an easier means of obtaining this end can be secured by providing two equal wheels in the change gear, so that there is but one position for the lever, or by arranging the change gear so that it can be disconnected altogether.

A careful inquiry as to the screwing feeds most frequently used indicates that for 6-in. centre lathes—Hendey-Norton type—75 per cent. of the time spent is on 9 to 16 threads per inch, and on 10-in. centre lathes, of the same type, a similar percentage on 6 to 12—inclusive—threads per inch. From this it seems that about six changes, arranged suitably to the particular size of lathe, are sufficient to meet the general demand, although it may be remarked that to employ a 10-in. centre lathe on such small work is not to use it to the best advantage.

This class of work, however, is more commonly done in this country on the hollow-spindle capstan or turret lathe, which is provided with a screwing head, as the spindles of the general-purpose screw-cutting lathes are solid, although it would add much to their efficiency, and little to the cost, if the spindles in the smaller machines were hollow. Since this is the case, the screwing work that usually comes to these lathes is of the square thread and worm variety, and the pitch is as often fractional as otherwise.

Now, it generally happens that it is necessary to cut a fine thread on the end of square-threaded screws to receive the check or lock nuts, and in the case of the Whitworth thread, where it has been cut by a single-pointed tool, it is usually finished by a chaser. Occasionally a small screwing head, adapted to fit the loose headstock spindle, is used for this purpose, or the turner sometimes prefers to do the whole operation with the chaser only.

These considerations, no doubt, account for the retention of the old system of change wheels in Britain, and those firms here of any note who have adopted the combined arrangement have confined themselves to as many changes as are necessary for the turning feed only.

Annexed are tables of Whitworth's and Sellers' standard screw threads. They are presented for the reader's perusal, in the event of his wishing to examine the validity of the foregoing remarks.

WHITWORTH STANDARD SCREW THREADS.

Diameter of screw.	Number of threads per inch.	Pitch in inches.	Diameter at bottom of thread.	Area at bottom of thread.
in.			in.	sq. in.
3/16	24	0·041	0·134	0·014
1/4	20	0·050	0·186	0·027
5/16	18	0·055	0·241	0·046
3/8	16	0·0625	0·295	0·068
7/16	14	0·071	0·346	0·094
1/2	12	0·083	0·393	0·121
9/16	12	0·083	0·455	0·155
5/8	11	0·091	0·508	0·204
11/16	11	0·091	0·571	0·256
3/4	10	0·100	0·622	0·304
13/16	10	0·100	0·684	0·367
7/8	9	0·111	0·733	0·422
15/16	9	0·111	0·795	0·496
1	8	0·125	0·840	0·554
1 1/8	7	0·143	0·942	0·697
1 1/4	7	0·143	1·067	0·894
1 3/8	6	0·166	1·162	1·058
1 1/2	6	0·166	1·287	1·299
1 5/8	5	0·200	1·369	1·472
1 3/4	5	0·200	1·494	1·753
1 7/8	4 1/2	0·222	1·590	1·986
2	4 1/2	0·222	1·715	2·311
2 1/4	4	0·250	1·930	2·926
2 1/2	4	0·250	2·180	3·733
2 3/4	3 1/2	0·285	2·384	4·464
3	3 1/2	0·285	2·634	5·450
3 1/4	3 1/4	0·307	2·856	6·402
3 1/2	3 1/4	0·307	3·106	7·563
3 3/4	3	0·333	3·323	8·673
4	3	0·333	3·573	10·027
4 1/4	2 7/8	0·348	3·804	11·365
4 1/2	2 7/8	0·348	4·054	12·908
4 3/4	2 3/4	0·364	4·284	14·404
5	2 3/4	0·364	4·534	16·146
5 1/2	2 5/8	0·381	5·012	19·720
6	2 1/2	0·400	5·487	23·640

WHITWORTH GAS THREADS

Size.	Diameter.	Diameter at bottom of thread.	Number of threads per inch.	Size.	Diameter.	Diameter at bottom of thread.	Number of threads per inch.
in.	in.	in.		in.	in.	in.	
⅛	0·3825	0·3367	28	1⅝	2·021	1·905	11
¼	0·518	0·4506	19	1¾	2·047	1·931	11
⅜	0·6563	0·5889	19	1⅞	2·245	2·1285	11
½	0·8257	0·7342	14				
⅝	0·9022	0·8107	14	2	2·347	2·2305	11
¾	1·041	0·9495	14	2¼	2·5875	2·471	11
⅞	1·189	1·0975	14	2½	3·0013	2·8845	11
				2¾	3·247	3·1305	11
1	1·309	1·1925	11	3	3·485	3·3685	11
1⅛	1·492	1·3755	11	3¼	3·6985	3·582	11
1¼	1·65	1·5335	11	3½	3·912	3·7955	11
1⅜	1·745	1·6285	11	3¾	4·1255	4·009	11
1½	1·8825	1·765	11	4	4 339	4·2225	11

UNITED STATES STANDARD SCREW THREADS

Diameter of screw.	Number of threads per inch.	Diameter at bottom of thread.	Width of flat.	Diameter of screw.	Number of threads per inch.	Diameter at bottom of thread.	Width of flat.
in.		in.	in.	in.		in.	in.
¼	20	0·185	0·0062	2	4½	1·712	0·0277
5/16	18	0·240	0·0074	2¼	4½	1·962	0·0277
⅜	16	0·294	0·0078	2½	4	2·176	0·0312
7/16	14	0·344	0·0089	2¾	4	2·426	0·0312
½	13	0·400	0·0096				
9/16	12	0·454	0·0104	3	3½	2·629	0·0357
⅝	11	0·507	0·0113	3¼	3½	2·879	0·0357
¾	10	0·620	0·0125	3½	3¼	3·100	0·0384
⅞	9	0·731	0·0138	3¾	3	3·317	0·0415
1	8	0·837	0·0156	4	3	3·567	0·0415
1⅛	7	0·940	0·0178	4¼	2⅞	3·798	0·0435
1¼	7	1·065	0·0178	4½	2¾	4·028	0·0454
1⅜	6	1·160	0·0208	4¾	2⅝	4·256	0·0476
1½	6	1·284	0·0208	5	2½	4·480	0·0500
1⅝	5½	1·389	0·0227	5¼	2½	4·730	0·0500
1¾	5	1·491	0·0250	5½	2⅜	4·953	0·0526
1⅞	5	1·616	0·0250	5¾	2⅜	5·203	0·0526
				6	2¼	5·423	0·0555

CHAPTER XXXI

FEEDING MECHANISM ON SADDLE

Saddle-feeding Mechanism.—The screw-cutting feed is usually effected by means of a nut, or portions thereof, in the saddle which engages with the leading screw. The screw generally rotates; motion being imparted to it after the manner already described, and no further gear exists between it and the nut which slides between suitable guides within the carriage apron, and is engaged or disengaged—closed or opened—with the screw by means of cams or excentrics, actuated by a conveniently placed handle. This motion must be positive, *i.e.* there must be no lost motion—a definite advance must be given to the saddle per revolution of the spindle. The regularity of the work and cut, and the comparative lightness of the latter, due to the delicacy of the tool and work, render any safety device to the mechanism unnecessary. In fact, all the parts should have abundance of strength to give rigidity and ensure accuracy. The turning feed mechanism is slightly more complicated, as the one-feed shaft—common British practice—is usually made to serve for the surfacing—cross or transverse—and traversing—sliding or longitudinal—feeds, the gearing for which is generally carried within the saddle apron, and so arranged that but one of these feeds—sliding, surfacing, or screw-cutting—can be engaged at one and the same time. Some of the larger lathes are designed to permit of taper turning by combining the surfacing and sliding or surfacing and screw-cutting motions. In the smaller machines, however, the majority of makers prefer to have all the feeds interlocking to avoid accidents, and to obtain the taper turning motion by some other means. For convenience, and to lessen the chance of injury either to the mechanism itself or to the work, several devices are employed, such as quick withdrawing motion to the tool; self-acting knock-out arrangements; arrangements whereby the tool is withdrawn simultaneously with the disengagement of the guide-screw nut, as when screw-cutting; the general disposition of the gear so that any of the feeds can be immediately disengaged at the saddle, and an automatic release to the turning feed when subjected to an undue load or shock—as when cutting irregular or intermittent work, etc. The old arrangement of belt feed previously described acted as a safety device instead of the latter attachment, but since the general adoption of positive feeds, such a device is necessary to avoid frequent breakdowns, particularly where the machines are continually worked to their full capacity. This arrangement is frequently fitted to the feed shaft—which is then in two portions—or in the saddle apron.

SAFETY DEVICES TO TURNING FEED GEAR

Amongst the safety devices for the turning feed gear, other than that already described, is that shown in Fig. 151. The design is similar to that adopted by Sir W. G. Armstrong, Whitworth and Co., Limited, Manchester, and fitted to their 18-in. centre high-speed lathe, shown at the Liège Exhibition. It is simply a Weston friction coupling, and connects the two sections K and L of the feed shaft. The outer casing M is keyed to K, and abuts against the shoulder of the shaft. Two or more keys are fitted in M to receive the phosphor bronze washers Q, so that they may rotate with the shaft K, whilst the steel plates P are keyed to the shaft L, yet are free to slide along it. A fine-threaded screw is formed on the latter shaft, upon which is fitted the adjusting and locking nuts O_1 and O_2. The plates P and Q are thrust together by the plate spring S, which is advanced and held in position by the nuts O_1 and O_2. The whole can thus be made to rotate as one piece, and is yet capable of slipping at any desired torque by regulating the compression of the spring. The casing M is filled with oil to ensure a regular slip, and prevent scoring of the plates when doing so. The end of the shaft K is chambered to provide a support to L.

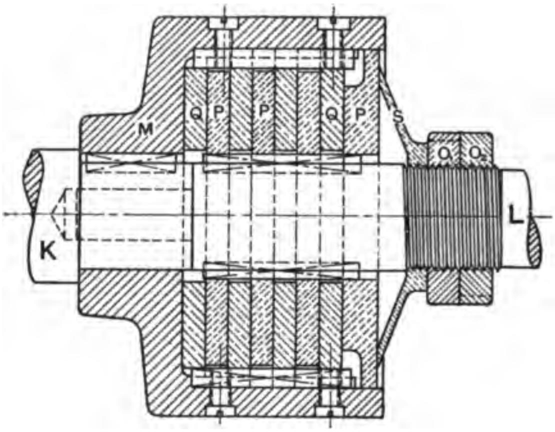


FIG. 151.—SAFETY CLUTCH.

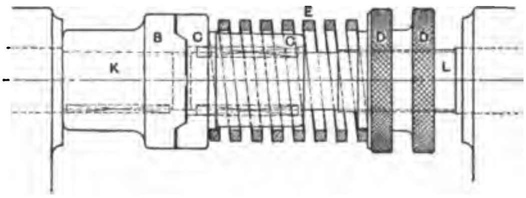


FIG. 152.—SAFETY CLUTCH.

Another design, also fitted to the feed shaft, is shown in Fig. 152, and takes the form of a quick-angled tooth clutch, after the manner of that used by Mr. Pearn in his patent tapping chuck. The feed shaft is in two sections—K, L—and upon the former is rigidly keyed the clutch B, whilst upon L is keyed the clutch C, which is capable of sliding along it. A fine-threaded screw is formed upon L to receive the adjusting and locking nuts D D. The spiral spring E, fitted between the clutch C and the nut D, keeps C engaged with B until the torque on the shaft is sufficient to overcome the initial compression of E when C is thrust back, L coming to rest while K continues to rotate. In both devices the springs are initially adjusted to carry the maximum load which the gear is designed to take, and only slip when this is exceeded. They are enclosed in casings to shield the parts from dirt and from being tampered with. Another method employed by a prominent maker to protect the gears consists of a simple flange keyed to each section of the shaft, and coupled together by a pin, through which the power is transmitted, the diameter of the pin being such that it will shear with any excess of load. Whilst the above are amongst the more recent schemes for this purpose, the old cone friction clutch is still very largely used, and when properly designed and constructed is thoroughly satisfactory. Unfortunately, however, it is more commonly placed in the saddle, where by the tightening of a nut or screw the cone is made to engage the sliding or surfacing feed gears, thus making the adjustment depend upon the discretion or muscular strength of the operator. No provision is made to limit the pressure between the cup and cone so that they will slip when wanted, beyond making the diameter of the nut or length of the lever small, so that the workman cannot nip them too hard together; yet this does not prevent him from using a spanner or hammer (which he generally does) in preference to his fingers to obtain the final "nip up." Under these circumstances the available torque is too indefinite to fulfil the function of a safety clutch.

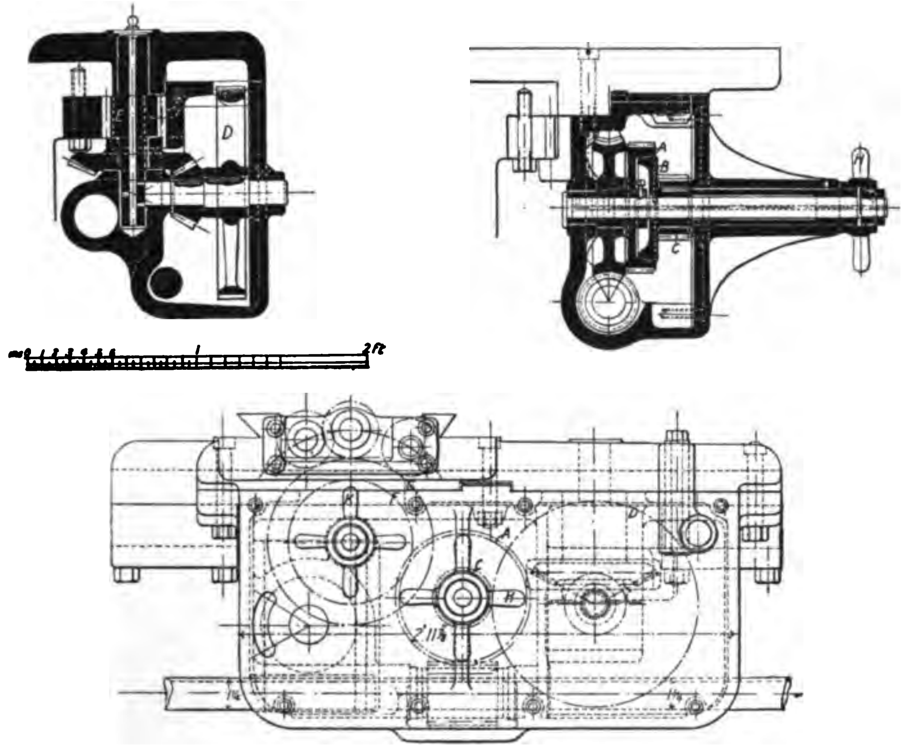


Fig. 153.—Saddle for Tangye Lathe.

Friction Cone Feed

The design of saddle apron—shown in Fig. 153—is for an 18-in. centre high-speed lathe, with duplex rests, by the Tangye Tool and Electric Company, Limited, Birmingham. Although friction cones are employed to engage the feeds, as described above, the feed-change gear at the end of the bed is driven by a belt, from a pulley on the fast headstock, running at twice the speed of the spindle, thus providing a safety link other than the cones. The worm on the feed shaft runs in an oil trough within the apron and meshes with the worm wheel keyed to the hub of the spur wheel A freely mounted on the transverse shaft. The friction cone B is provided with a long sleeve, which extends beyond the bearing in the apron upon which the pinion C is keyed. The end of the sleeve and collar pinned to the end of the transverse shaft are screwed to receive the nut H. These screws are of different pitches of thread, so that by turning H the cone B is engaged with, or withdrawn from, A. The transverse shaft carrying the wheels F and G is similarly fitted. Thus, when surfacing, K is locked, and the motion is transmitted through the wheels A, F, and G—A and B being disengaged—to the pinions on the ends of the surfacing screws—of which there are two, one for each set of rests—in the saddle. When sliding, K is unlocked whilst H is locked to engage A and B, the motion being transmitted through C, D, and the bevel pinion and wheel to the rack pinion, which is combined with the shaft and meshes with the vertical rack. It will be observed that the arrangement and construction of the gears and apron permit of all the wheels being carried between bearings. Rigidity and regularity of motion are thus secured.

Positive Feed

In the saddle for an 8½-in. centre lathe, by Messrs. Greenwood and Batley, Limited, Leeds, shown in Fig. 154, the surfacing and sliding feeds are positive and the mechanisms entirely independent. (For feed-change gear see Fig. 140.) On the longitudinal feed shaft at the front of the bed and within the apron is keyed a broad-faced pinion, which engages with the two spur pinions formed on the end of the worm shaft in the drop-worm brackets. These brackets swivel about the axis of the feed shaft, and, when the worms are engaged with their wheels, are held in position by trigger attachments, for which provision is made in the apron. A ledge is also fitted to the apron, upon which the worm-box rests when withdrawn. When traversing, the single-threaded worm on the right-hand side of the saddle is engaged with the worm wheel—forty teeth, ½-in. pitch—keyed to the hub of the pinion; whilst the latter engages with the wheel on the rack-pinion shaft. The rack pinion is formed on the end of the shaft, and is supported immediately under the rack, the bearing being cut away at the top to clear it. A hand-traversing motion is obtained through the pinion on the extreme right of the saddle meshing with the wheel on the rack-pinion shaft. The surfacing motion is secured through the double-threaded worm on the left of the saddle, which engages with the wheel—forty-eight teeth, ⅜-in. pitch—on the boss of the spur pinion, and an intermediate wheel meshes with the latter and the pinion on and between the bearings of the surfacing screw. The outer end of this screw passes through another with a much quicker pitched thread, and is free to rotate but tied transversely on the same, to permit of the operator rapidly withdrawing the tool—and rests—when cutting a screw to a blind end. The quick-pitched screw has its nut in the outer bearing, and the hand lever formed on its end abuts against the stop pins on the carriage, so that the rests and tool can be returned to their former position. It will be observed that the fine adjustment to the tool by the surfacing screw and hand wheel is in no way disturbed by the quick withdrawing mechanism.

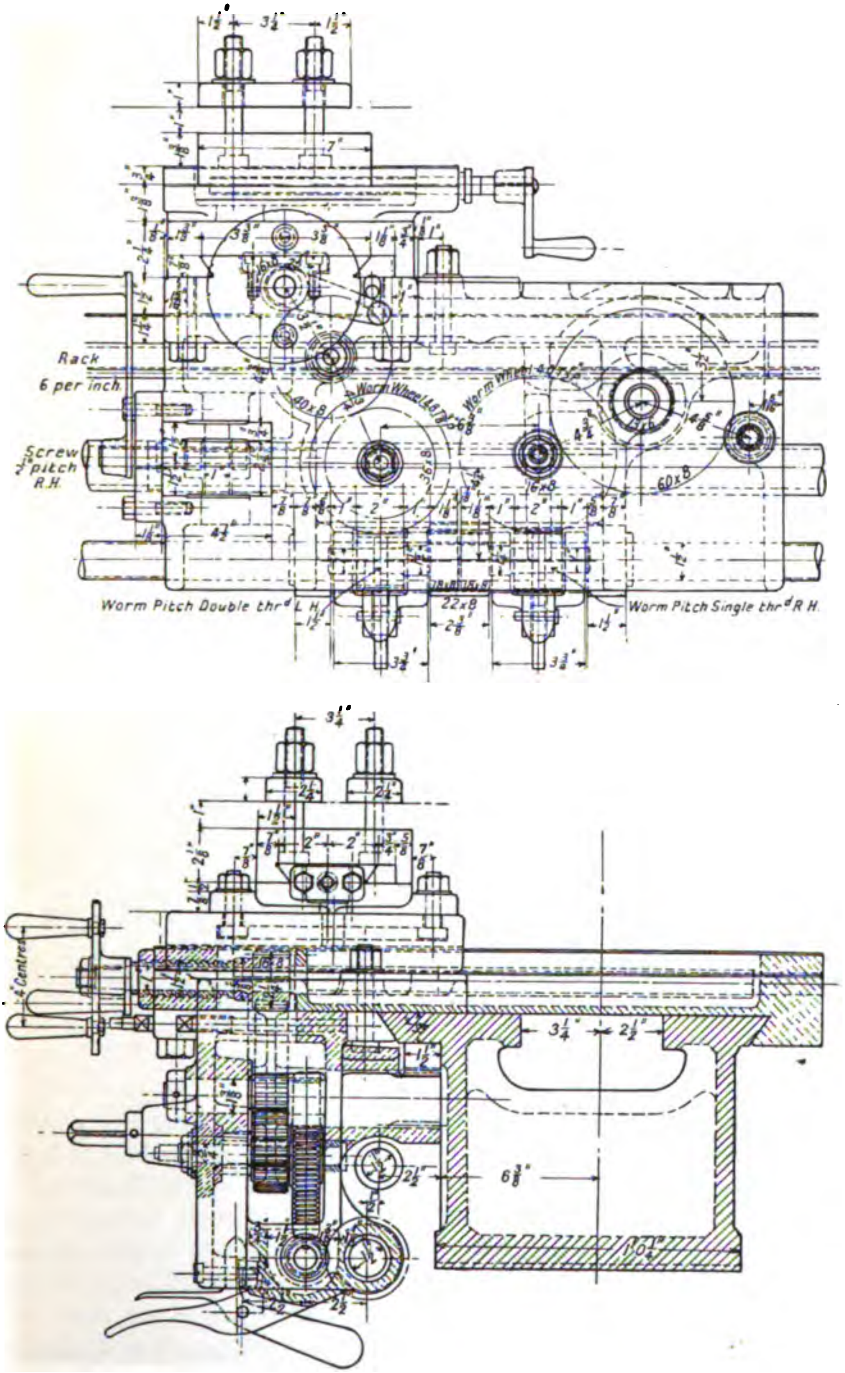


Fig. 154.—Saddle for Greenwood and Batley Lathe.

A design for a similar size of lathe by John Lang and Sons, Johnstone, is shown in Fig. 155. (The change-feed mechanism for this machine is shown in Fig. 144). The worm on the feed shaft is carried in an oil bath within the saddle apron, and meshes with the wheel on the hollow shaft S, whereon are freely mounted the spur wheels V and Y. The internal shaft T carries the push rod with the cotter, and has a friction clutch keyed to its outer end. An annular space within the hubs of V and Y enables the cotter to rotate independently of the wheels. To engage the sliding feed the handle nut at the end of the shaft T is tightened, thereby locking the clutch on same and worm wheel together. The push rod is then thrust back, when the cotter enters the keyways of and engages the pinion V, which meshes with the wheel W on the rack pinion shaft, the pinion and shaft being formed in one piece. For the surfacing feed the cotter is made to engage the outer wheel Y, which gears with the pinion A on the end of the surfacing screw through the intermediate wheel Z. A hand-wheel keyed to the stem of the pinion X, which meshes with the wheel W on the rack pinion shaft, furnishes a hand traverse motion. The guide screw nut is in halves, and is fitted between the cheeks on the apron. Excentrics formed on the end of the small transverse shaft, carrying the hand lever, enter each half nut, which, on the turning of the lever, engages—or *vice versâ*—with the screw; a spring

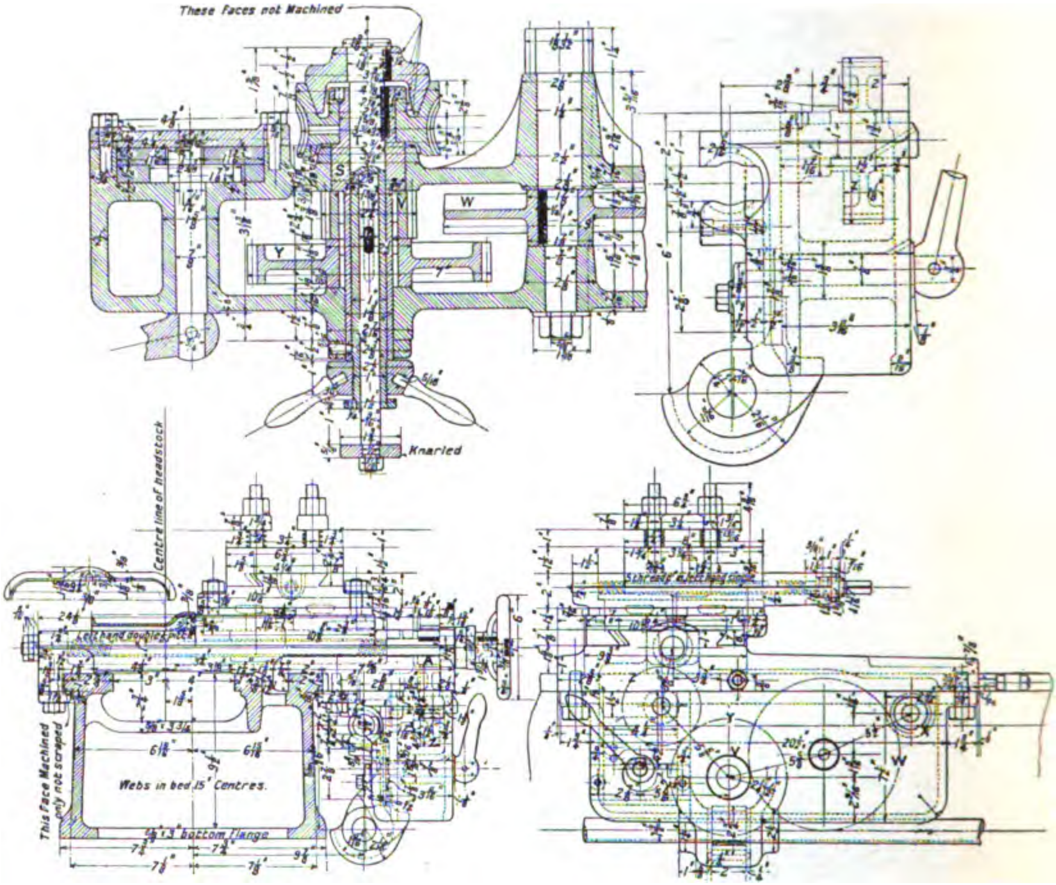


FIG. 155.—SADDLE BY JOHN LANG AND SONS.

pin and plunger assisting to keep them in position. By referring to the feed-change gear mentioned above, it will be clear that when the guide screw is rotating the feed shaft is at rest, and as the cotter can only be engaged with the wheel V or Y, it is impossible for more than one of these feeds to be in action at one and the same time. No quick withdrawing motion to the tool is provided, but the surfacing screw being of double thread and large pitch serves the purpose for the ordinary run of work.

PARTICULARS OF WHEELS

Wheel.	Teeth.	Pitch.	O.S.D.	P.D.	Centres.	Bore.	Material.
						in.	
V	20	8 D.P.	2¾	2½	5⅝	1⅜	M.S.
W	70	8 ”	9	8¾	5½	1	C.I.
X	18	8 ”	2½	2¼		—	M.S.
Y	54	8 ”	7	6¾	5⅝	1⅜	C.I.
Z	36	8 ”	4¾	4½	3⅜	1	”
A	18	8 ”	2½	2¼		1	M.S.
Rack pinion	14	⅜ C.P.	1⅞ $\frac{1}{32}$	1⅝ $\frac{1}{32}$ $\frac{1}{64}$	—	—	”
Worm wheel	36	$\frac{5}{16}$ ”	3¾ $\frac{1}{32}$	3$\frac{9}{16}$ $\frac{1}{64}$	3$\frac{1}{16}$	1⅝	C.I.
Worm . .	L.H.S.T.	$\frac{5}{16}$ ”	2¾	2½ $\frac{1}{32}$ $\frac{1}{64}$		1⅜	M.S.

Several arrangements have lately been devised to prevent the simultaneous action of more than one feed. If the mechanism is to be fool-proof, however, not only must the other feeds remain inactive, but they should be incapable of engagement whilst any part of another feed mechanism remains in gear. The majority of designs provide for the disconnection of the screw-cutting gear with the engagement of that for the turning feeds or motion to the feed shaft. Clearly this is insufficient, as it allows of the traversing feed being put in action whilst the nut continues to embrace the screw; or the mechanism in the saddle for the former may be in gear; and although the feed shaft is at rest, the screw-cutting gear is open to connection. The surfacing feed mechanism is subject to the same fault, although in a lesser degree, as the engaging device for this feed is also usually made to serve for the traversing feed, and alternately couples the one or other. Yet it is generally possible when screw-cutting for the surfacing feed to be actuated through the rack and pinion gear. To control all the feeds from the saddle and make them entirely interlocking presents but little difficulty, and need not incur much expense.

FRICTION CONE FEED WITH INTERLOCKING TURNING AND SCREW-CUTTING GEAR

In the design by the Lodge and Shipley Company, shown in Figs. 156 and 157, the screw-cutting and traversing feed gear is arranged in such a manner that the possibility of their simultaneous action is avoided. Motion is transmitted to the screw by the gear at the end of the headstock—see Fig. 147—and the double-clasp nut is made to close upon the same by means of a cam actuated by a lever at the front of the apron. The feed shaft commonly used in British lathes is here dispensed with, and the work of driving the turning feed gear is performed by the guide screw. This screw is furnished with a long keyway to receive a key fitted in the sliding muff carrying the bevel pinions, either of which can be made to engage the wheel—depending on the direction of feed desired—by means of a lever, rod, and claw, operated from the front of the saddle. The sliding and surfacing feeds are independent, are engaged by means of friction cones, locked by knurled nuts outside the apron, and driven from the spur pinions on the bevel-wheel shaft. The apron is tongued and bolted to the carriage. The inner wing, which furnishes a bearing to the rack-pinion shaft, etc., is supported and slides on the steel strip let into the bed to reduce the springing between the rack and pinion

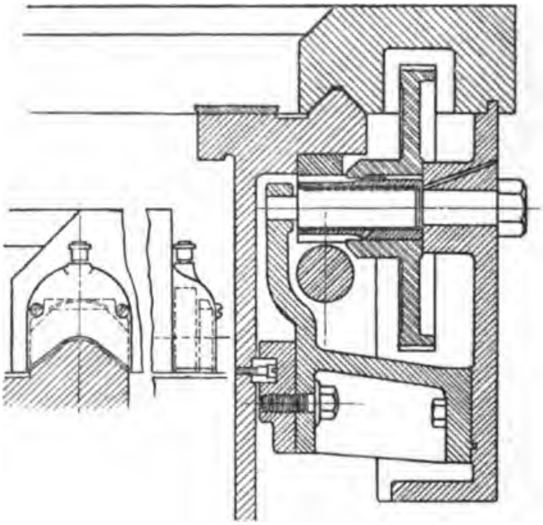


Fig. 157.—Section of Apron.

when heavy cutting—see Fig. 157. To ensure accuracy of fit and reduce cross-winding the guide screw-nut is chased from the solid and fitted directly to the slide, no adjusting strips or screws being used. Notwithstanding the fact that the key in the leading screw extends the full length of the bevel pinion muff, it does not prevent the burring of the screw threads and consequent destruction of the nut. This method, although at one time common in American machines, has been gradually dropped, and the more recent productions are fitted with a feed shaft. It will be observed that a stop finger is fitted on the end of the rod carrying the claw, which prevents the nut closing upon the screw, when the sliding or surfacing feeds are in use, and that the recess in the nut casing to receive the finger prevents the engagement of the latter feeds when the nut is closed. It is, however, still possible for these—sliding and surfacing—feed gears to be locked and motion transmitted from the rack and pinion when screw cutting; yet the friction cones, being situated so near the work and locked only by the knurled nuts, and not by a screw-key, may be sufficient to avert disaster if accidentally coupled.

The design of saddle adopted by de Fries and Co., Düsseldorf, is shown in Fig. 158. It is fitted with a short screw or worm L, which engages the long segmental nut or rack T bolted to the bed, and takes the place of the usual guide screw and nut. The feed shaft Z derives its motion from the spindle through the reversing and change gear at the rear of the fast headstock—see Fig. 133. The sliding and surfacing feeds are obtained through the worm H, which is mounted freely on Z and coupled to it by means of the clutch N. The worm meshes with the wheel I on the transverse shaft, whereon is keyed the friction clutch J. On the end of J is fitted a screwed cylinder, and the handle nut R on the same moves the clutch transversely, so that it may engage either E or U, which are free on the shaft. When surfacing J is engaged with E, which meshes with the wheel F keyed

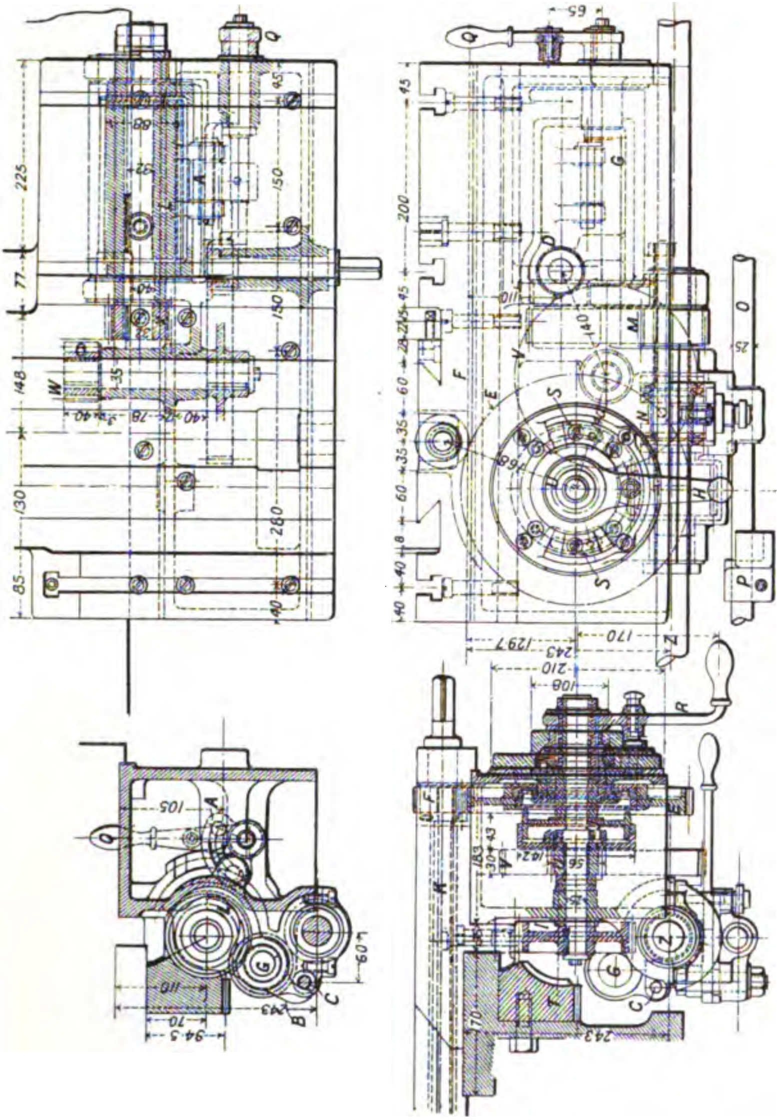


Fig. 158.—De Fries' Saddle.

to the surfacing screw. The sliding motion is obtained by engaging J with the pinion U, which gears with V on the rack pinion shaft; the rack pinion W meshes with the rack on the under side of the segmental nut T. Two stops S are fitted to the bracket on the apron front to prevent the possible jamming of the clutch J within the cup by the handle nut R, and also to allow for adjustment and wear. The longitudinal adjustment to the saddle is obtained through the hand-wheel and pinion D, which meshes with V on the rack pinion shaft. When screw-cutting the clutch N is engaged with the wheel M, which drives wheel X on the end of the short screw L through an intermediate wheel. The screw L is carried in an oil-box swinging about the centre G, and the handle Q actuating the cam A engages or disengages L with T. A lip B is cast on the swinging bracket and fits between the bearing and the collar on the rod C when the clutch N is in mid-position. As the rod C is controlled by N it is impossible to engage N with H whilst L and T are in mesh. Similarly, when N and H are in mesh the screw L cannot be engaged with T. In this way the simultaneous engagement of the sliding and screw-cutting feed mechanisms is prevented. The clutch N can be automatically disengaged from the gear M or worm H by means of the stops P on the rod O. By this arrangement the tendency to cross-thread when cutting multiple-threaded screws is considerably reduced.

It is difficult to see what advantage is gained by the substitution of the long nut for the guide screw, except that the former is more rigid than the latter. It is doubtful, however, if this nut can be reproduced with as much accuracy as the screw, and it is certainly much more expensive to make. If the screw be made sufficiently large in diameter, a degree of accuracy sufficient for all practical purposes can easily be obtained.

POSITIVE-FEEDING MECHANISM WITH INTERLOCKING TURNING AND SCREWING FEEDS

In the arrangement by William Sellers and Co.—shown in Figs. 159 and 160—the possibility of simultaneous engagement of the screw-cutting and traversing feed is altogether avoided. Two bevel pinions are keyed to the longitudinal feed shaft, but are free to slide along the same, and either of them can be made to engage with the bevel wheel on the diagonal worm shaft by means of a rod and counterbalanced lever at the front of the carriage. The worm is multi-threaded, and its angle, together with that of its axis, is such as to allow the wheel to slide through the worm. This worm wheel C is provided with a clutch face on each side of its hub, to engage with the corresponding clutches on the pinion A at the inside for the traversing or sliding motion, and with the hub of the outer wheel B for the surfacing motion—as shown in the figure the wheel C is engaged with B_1. The wheel B meshes with the combined pinion and rotary nut F, which is only free to rotate when the surfacing feed is in use. To engage C with B_1 or A, an excentric bush is fitted on the end of the short transverse shaft carrying the wheel C. It receives the small crank shaft within the bracket, which is operated by a handle. When C is advanced to mesh with A for the traversing feed, the pawl E simultaneously engages with the teeth of the pinion F on the surfacing screw, thus positively locking and preventing the possible engagement of these two feeds at one and the same time. A similar locking arrangement is provided for the screw-cutting and traversing feed by the projecting lip D on the guide-screw nut push rod, which prevents the engagement of C with A whilst the nut embraces the screw, or *vice versâ*.

It will be observed that, with this arrangement, there is no possibility of accident to the traversing feed gear by the rack pinion driving backwards when the screw-cutting mechanism is in use. A taper-turning motion is obtained by the simultaneous engagement of the screwing and surfacing feeds. The guide screw is placed within the bed to reduce cross wind of the saddle, and is protected by the upper front way from dirt, and supported throughout its entire length by a lower ledge cast with the bed.

FINE HAND-TRAVERSING MECHANISM

An outstanding feature of the American saddle is its handiness for rapid and accurate manipulation. In the early machines and until recently the American firms did not, as a rule, supply a swivel slide to the rest, and, as a consequence, had to furnish a fine hand-traversing motion in the saddle. This has proved so convenient that it has been retained in the machines of to-day, even when a swivel slide is provided. The American operator prefers it to the slide for hand traversing, and in the smaller machines it is generally admitted to be the better method. The British firms, as a rule, have always furnished a swivel slide to the rests and a quick hand-traversing motion to the saddle. In the older machines this last motion was only intended for bringing the tool quickly up to the work, and not for a hand feed. The pinions and racks were cast, thus producing an irregular or jerky motion altogether unsuitable for fine hand traversing. The latter was secured through the swivel slide screw. When this was the function of the hand-traverse motion, the single or double lever was usually fitted directly to the rack-pinion shaft in the smaller lathes, and with only sufficient gear purchase in the larger machines to enable the operator to move the carriage along

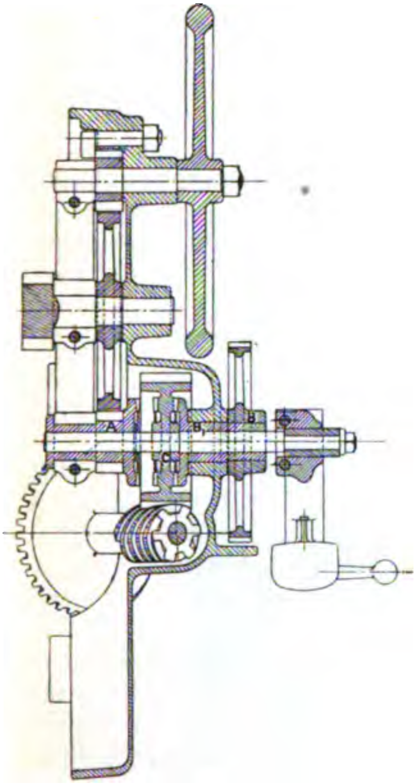


FIG. 159.—SELLERS' APRON.

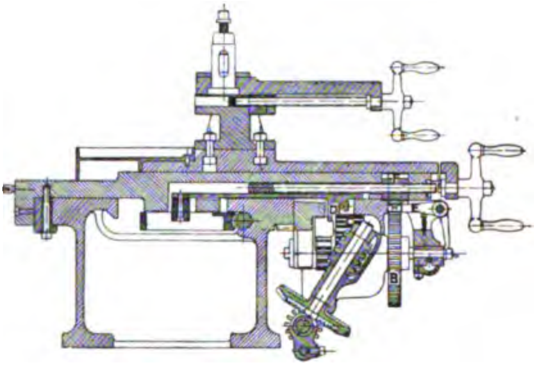


FIG. 160.—SELLERS' SADDLE.

the bed. The perfection of the wheel-cutting machine, and the common use of machine-cut gears, together with the change of practice—partly brought about by the American design of saddle—have all led up to the present design, which leaves little to be desired for fineness of adjustment and regularity of motion. A few makers, however, adhere to the single or double lever, which does not give the machineman the same opportunity of observing the work whilst hand feeding as he would have with a hand wheel. When taking a heavy cut with the power-traverse feed, however, the motion is not so continuous as is desirable, even in some comparatively new machines. In the more recent lathes by British makers, the racks and pinions are of steel—machine-cut—permitting the use of a finer pitch, and the parts within the apron have been strengthened to reduce springing, whilst in some an outer bearing or support to the rack pinion immediately under the rack has been provided. These modifications considerably reduce the above objection, but the smallness of the pinion diameter is detrimental to uniformity of motion.

In a recent design of high-speed lathe, by J. Buckton and Co., Leeds, shown in Fig. 161, the usual rack pinion has been displaced by a spiral pinion, or multi-threaded worm, which meshes with the vertically straight tooth rack on the side of the bed. Motion is transmitted from the feed-change gear below the headstock to the longitudinal feed shaft, and suitable gearing within the apron communicates with a pinion shaft whose axis is inclined to an angle of about 30° to the rack, after the manner of the Sellers' planing machine drive. Four teeth of the pinion are continually in gear, so that in the 14-in. lathe, *e.g.*, a ¼-in. cut with ¼-in traverse on mild steel at 60 ft. per minute can be taken without any perceptible vibration. Both the sliding and surfacing have their reversing mechanism in the saddle, and a quick hand motion for operating the saddle in either direction is provided. The screw-cutting motion, when fitted, consists of a leading screw situated at the front of the bed, actuated in the usual way by change wheels at the end of the headstock.

Taper-turning Attachments.—As already mentioned in the description of saddle mechanisms, a taper-turning feed can be secured by combining the longitudinal and cross-feeds. The feeds obtained by this means are limited by the values of the change wheels usually supplied and the fixed train of gear on the saddle apron, so that it is not always possible to get the feed desired. This is particularly the case with small lathes where the taper work is very varied. Such tapers, *e.g.*, as the Morse standards—which are approximately 1 in 20—could not be cut by this method. The fault here lies, of course, with the tapers which, in the case of the Morse, could be just as easily 1 in 20 as almost that.

Again, in small work it generally happens that only a portion of the job is tapered and the remainder parallel, so that it is necessary, for economical production, to be able to change quickly from taper to parallel turning, or *vice versâ*. This may be accomplished in several ways, but the most satisfactory method for moderate taper is that of the adjustable bar or slide at the back of the lathe.

An attachment of this description, by John Lang and Sons, is shown in Fig. 162. The tapering bar A is supported throughout its entire length by the guide bar B, and swivels about a pin fixed in the centre of the same. To

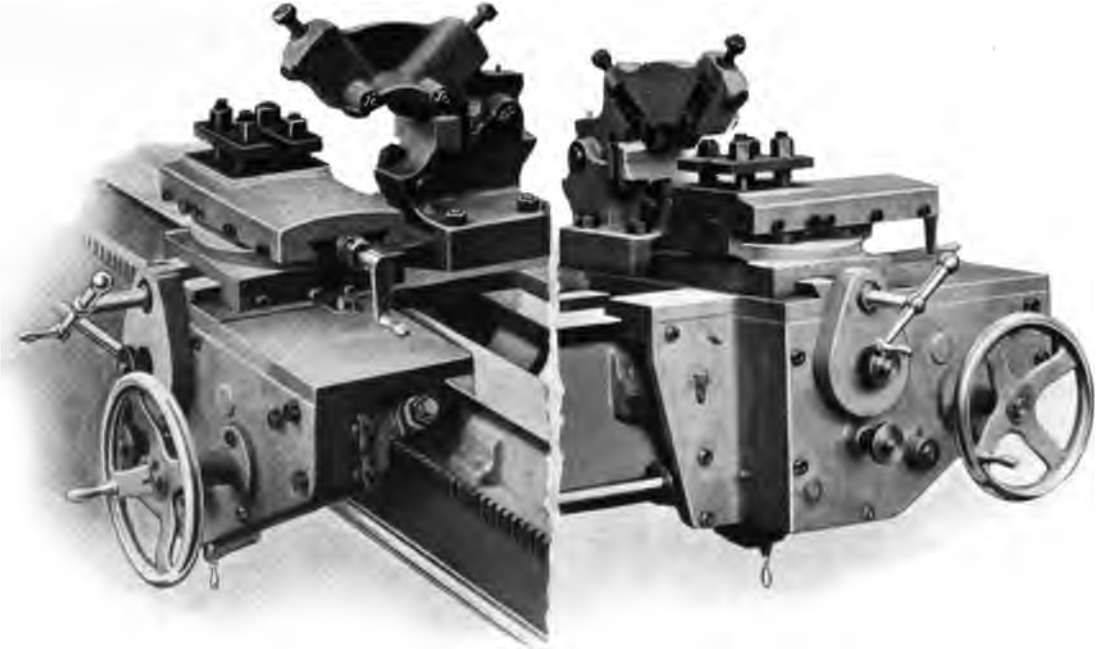


FIG. 161.—SADDLE AND RESTS FOR 12-INCH CENTRE LATHE (BUCKTON).

facilitate setting the end of the guide bar nearest to the fast headstock is graduated in degrees, and A has a corresponding zero mark. The bar B is carried by the bracket C bolted to the rear of the saddle, so that it travels with it, whilst B is held by the stud D in the bracket E, which can be locked

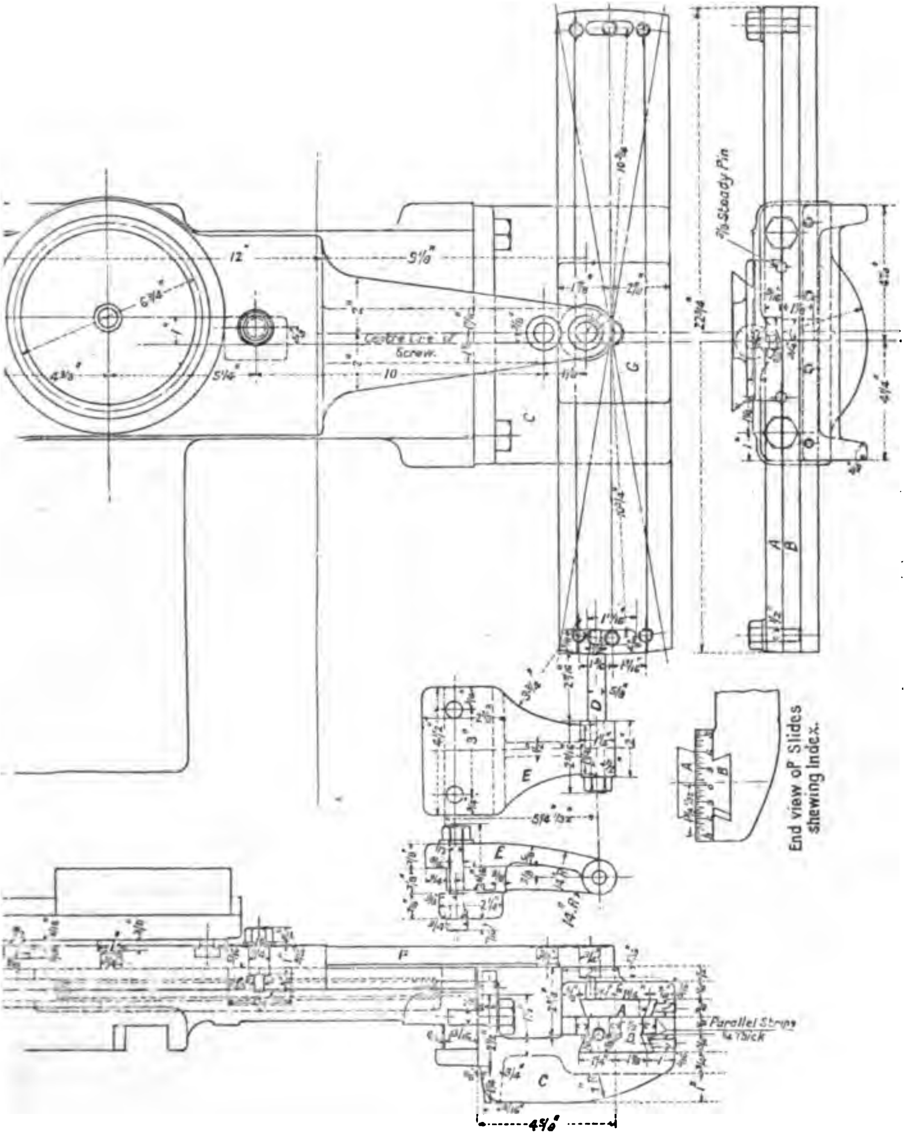


Fig. 162.—Tapering Attachment.

to the bed at any desired position. When taper turning, the surfacing screw nut is disengaged, and the projecting arm F on the bottom slide is coupled to G on A, so that as the saddle is advanced along the bed, the slides are compelled to follow the tapering bar. A cross adjustment to the tool when so engaged is obtained by setting the swivel slide, as shown in figure. The arrangement permits of a quick and easy change, and of obtaining a taper anywhere within the length of the bed.

CHAPTER XXXII

GENERAL DISCUSSION OF SCREW-CUTTING AND CROSS-THREADING

In some of the more recent machines the old time $\frac{1}{2}$-in. pitch guide screw or leader has been displaced by one of a finer pitch. The reason for this is not at all clear. Some engineers have the idea that a fine-pitch screw, like a small-pitch wheel, gives a more uniform motion, and whether this, or the impression that a fine-threaded screw looks neater and more in proportion to the other parts of the machine in small lathes, is responsible for the departure, it is difficult to say. It is a common practice in America, in the smaller sizes of lathes, to furnish a leading screw having a pitch of 5, 6, or 7 threads per inch. This is quite in accord with their adopted system of pitch, viz. so many threads per inch. Providing screws of the usual number of threads or pitch can be cut as easily with a fine-pitch leader as with a coarse, it does not matter which is adopted. This, however, is not always the case, as will be seen from the following comparison.

Taking a lathe where the leader is driven from the spindle through a train of change wheels mounted on a quadrant at the rear of the headstock, the nut being withdrawn when the tool has reached the end of the screw, and the saddle brought back by hand, we have:—

Case (*a*): **Single Thread Even-pitched Screws.**—When cutting a single-threaded screw of an even number of threads per inch with a leader of two threads per inch, there is no possibility of crossing or splitting the thread—providing, of course, the relative position of the tool and nut remain undisturbed—as it will be a multiple of the leader, and as the nut can only enter every $\frac{1}{2}$ in. or revolution of the leader, the tool will always be opposite the groove of the screw being cut. Should the leader have a pitch of 6 threads per inch no trouble will be experienced so long as the screw operated upon is a multiple of the same, viz. 6, 12, 18, etc., threads per inch. As the nut can be engaged every $\frac{1}{6}$ of an inch, there is a chance of cross-threading at two distinct places, viz. $\frac{1}{6}$ and $\frac{1}{3}$ of an inch when cutting screws of any other even number, such as 2, 4, 8, etc., threads per inch. Similarly an even-pitch screw can be cut by a 7 per inch leader if it is a multiple of the latter without interference, but for any other even pitch the chances of crossing the thread are increased to 6 out of 7 times.

Method (1).—An obvious way of surmounting this difficulty would be to provide a reverse motion on the countershaft wherewith to reverse the machine without disengaging the nut and leader.

Method (2).—Such a method, however, is slow and tedious, and the operator generally prefers to disengage the nut and run the carriage quickly back by hand to the starting point. When this method is adopted, the nut is engaged with the leader prior to taking the first cut on the screw. The faceplate and wheel on the end of the leader are then marked in line with some fixed object—frequently a chalk mark on the bed. A stop is brought up to the saddle against which it (the saddle) may abut, as it is essential that the saddle should return to the same position each time. After the cut has been taken, and the saddle has been brought against the stop, the nut is engaged with the leader when the marks on the faceplate and wheel come simultaneously opposite the fixed mark. Here the operator may require to stop the lathe to bring the marks in line, but unless the leader be of a very fine or odd pitch it is not generally necessary. Now, if the carriage is not brought to its original position, it is possible to engage the nut when the marks come simultaneously together, and yet cross-thread the screw. The result will be the same if the saddle is brought against the stop, and the nut engaged prior to the marks coming together.

Case (*b*): **Single Thread Odd-pitched Screws.**—If the screw operated upon contains an odd number of threads per inch, it is possible to split the same with a 2 per inch leader, as the nut can be engaged every ½ in. when the tool is opposite a half thread on the screw. With a 6 per inch leader there is a likelihood of cross-threading in five places when cutting 1, 5, 7, 11, etc., threads per inch, but in the case of 3, 9, 15, etc., it will only halve the thread. Obviously a 7 per inch leader will cut 7, 21, 35, etc., threads without cross-threading, as they are multiples of the same, but with any other odd pitch the likelihood of crossing the thread is correspondingly increased. Briefly stated, the fraction of the leader in inches corresponding to the revolutions made by same, multiplied by the number of threads being cut, must equal a whole number to avoid interference. Thus, with a 6 per inch leader and 9-threaded screw:—

Inches corresponding to revolutions of leader.	Number of threads in screw.	
$\frac{1}{6}$	× 9	= 1½ (crossing)
$\frac{2}{6}$	× 9	= 3
$\frac{3}{6}$	× 9	= 4½ (crossing)
$\frac{4}{6}$	× 9	= 6
$\frac{5}{6}$	× 9	= 7½ (crossing)

When cutting this class of thread with a 2 per inch leader, and adopting method (2), the task is comparatively easy, as the marks come together every second revolution of the leader, whilst they are diametrically opposite at the intervening revolution. With a 6 per inch leader the operation is by no means so simple, since the marks only come together every six revolutions, and there is not the distinct difference between them in the intervening revolutions, as in the preceding case. The occasions when the marks are diametrically apart are when the screw operated upon contains such thread as 3, 9, 15, etc., per inch, in which case the nut may be engaged every second revolution of the leader.

The marks at times may be less than one-seventh of a revolution apart, in which case, when running quickly, as when cutting a screw of small diameter, it is most difficult, and sometimes next to impossible, to catch them. Under these circumstances the operator would stop the lathe to bring the marks together, and, having run the saddle back to the stop, would enter the nut before re-starting the machine. The loss of time occasioned by the stopping and starting at each cut, and the absence of the man from the tool at the commencement of the cut—as it is not common nowadays to furnish a belt-striking gear which can be controlled from the saddle—are serious objections to this system. If the machine were fitted to allow of method (1) being adopted there would be no need to trouble about the marks on the faceplate and wheel on the leader; neither would it be necessary to bring the saddle back to the original starting place each time, as the relative position of the nut and screw are always maintained. While this method is slow, it is sure and simple; but the additional belt and pulleys on the line and countershaft are against its adoption, and unless the machine can be reversed from the saddle, it has the objection mentioned above.

Method (3).—If the lathe is provided with a reversing train to the feeding mechanism similar to those shown in Figs. 116 *ante*, and the clutch actuated from the saddle either by hand or a self-acting knockout device (obtained by fitting adjustable tappets on the rod), another and more positive means of preventing cross-threading may be secured.

The spur wheels on the end of the spindle and change wheel or reversing shaft have equal numbers of teeth and the clutch can only engage the wheel at one place, the relation between the spindle and shaft being always maintained. When the tool has reached the end of the screw the clutch is first disengaged, thereby bringing the change wheels and leader to rest, whilst the work continues to rotate; and the nut is afterwards withdrawn from the leader. If the saddle be then moved back so that the length of the screw or screw produced (*i.e.* the travel of the tool) is a definite number of inches, the screw and leader will each contain an exact number of full threads, and the nut will correctly engage with the leader. After entering the nut the clutch may be engaged, and the same process repeated. Thus, by always stopping and starting at the same place interference is avoided. The arrangement conveniently lends itself to both operations (*i.e.* disengaging first the clutch and then the nut, and *vice versâ*) being performed by one movement, as a small lever on the actuating rod, and travelling with the saddle, can be connected by a link to the nut handle.

Method (4).—If the length of the screw be short, it may be quicker to engage the clutch with the opposite wheel, thereby reversing the feeding mechanism without disengaging the nut, in which case the tool will always be in the correct position, no matter where the cut ends or starts, or whether the number of threads be even, odd, or fractional.

It is evident that the above result cannot be obtained when the clutch is mounted on the leader.

The fine-pitch guide screw need not be further discussed. Sufficient has been said about its accompanying disadvantages to preclude its adoption by preference, unless provision be made similar to that referred to in method (3). In what follows only $\frac{1}{2}$-in. pitch guide screws are considered.

Case (*c*): **Single-thread Fractional Pitched Screws.**—The case of a fractional pitched screw, such as a $\frac{3}{4}$-in. pitch or $1\frac{1}{3}$ threads per inch, is similar to that last discussed (case *b*), in so far that if method (3) be used the saddle must always return to the starting place, and the nut only be engaged when the marks come simultaneously together. This, however, will only occur every third revolution of the leader, so that the chances of crossing are increased to two out of three. If method (3) be used, the length of the screw—or screw produced—may be in inches or half-inches, provided it is a multiple of $\frac{3}{2}$—the shortest length in which the leader and screw contain a full number of threads—or the number of threads contained therein may be divisible by 2. Similarly, with this system 3, when cutting a screw of $1\frac{3}{4}$-in. pitch or $\frac{4}{7}$ threads per inch, the necessary length of screw for proper engagement must be a multiple of $3\frac{1}{2}$ or the number of threads contained therein are divisible by 2. Whilst with method (2) there is no need to have a full number of threads or inches in the screw, the difficulty of catching the marks is considerably increased, as a correct engagement is only secured every seventh turn of the leader, and the marks in the interval may be but $\frac{1}{7}$th of a revolution apart. It is this class of work which gives the operator most trouble, involving harassing calculations on the lathe bed or 12-in. rule, and causes the greatest loss of his time. It appears that method (4), which is simple and positive, would furnish an entirely satisfactory solution of all cases if the reversing mechanism to same were fitted with a quick return train, capable of being controlled from the saddle. In this way the time spent in returning at the same rate as the forward feed is saved. It may be objected to this arrangement that there will be increased wear of the leader and nut. There seems, however, no reason why the diameter of the former and the length of the latter should not be enlarged to allow for same.

Method (5).—Messrs. Archdale and Co., Birmingham, effectively overcome the above difficulties in their small lathes by providing interchangeable nuts and small hollow leaders of different pitches. These can be slipped on and connected to a permanent spindle—occupying the position on the machine usually assigned to the guide screw—driven by a fixed train of wheels at the rear of the headstock, and by this means the wear is distributed over a number of pieces which can be easily and cheaply replaced. The system is, however, only applicable to small work.

Case (*d*): **Multiple Thread Screws.**—In a screw having more than one thread, each thread may be treated separately, so that what has already been said regarding the various methods, pitches, or number of threads per inch above applies equally to this case. Thus a $\frac{1}{2}$-in. pitch, double-thread screw may be treated as a $\frac{1}{2}$-in. pitch single, and a $\frac{3}{4}$-in. pitch treble as a $\frac{3}{4}$-in. pitch single thread, and so on. The width of tool and depth of thread must, of course, be proportioned to suit. The question of the sub-division of the screw arises, however, and this may be performed in several different ways, depending on the pitch and number of threads. If, *e.g.*, the pitch of the screw operated upon is a whole number of inches—*i.e.* 1-in., 2-in., 3-in., etc., pitch—and double-threaded, the operator would probably adopt method (2), but would prefer to mark the faceplate in two places—diametrically opposite each other. The nut would then be engaged as these marks came together alternately with those on the leader wheel and bed, and a cut would be taken down each thread in succession before altering the feed or depth of cut. Similarly with a treble-threaded screw of $1\frac{1}{2}$-in., 3-in., $4\frac{1}{2}$-in. pitch, the faceplate would be divided into three parts and marked; a cut being on each thread in succession, as described above.

For the majority of multiple-threaded screws, however, the above system is inapplicable, and a method which will meet all cases is the following: Assuming the change-wheel shaft to rotate at the same rate as the spindle—which it generally does—the operator would select, as a member of the train, a first-motion change wheel, having its number of teeth a multiple of the number of threads to be cut. Having engaged the nut, fixed the stop behind the carriage, and marked the faceplate and wheel on the end of the leader in line with the fixed mark prior to taking the first cut as in method (2), he would then divide the teeth of the first-motion change wheel into the required number of parts, making sure that the first marked tooth was diametrically opposite a space in the driven wheel. Proceeding as in method (2), he would then complete one thread. To cut the second thread the first-motion change wheel should be withdrawn—clear of the driven wheel, so that the remainder of the train and leader are undisturbed—and turned round with the spindle until the next divided tooth came opposite the space vacated by the first marked tooth, and would be engaged with the same. As the mark on the faceplate has changed its position during this operation, it is necessary to re-mark the same, and to save confusion the first mark should be obliterated.

The result of this operation is that the work with the spindle has been simply turned round to the next thread without disturbing the change wheels, guide screw, or position of the tool. A similar result would be obtained by dividing the faceplate into the number of parts corresponding to the threads, and having disengaged the change wheels, moving it—the faceplate—round one division. This, and all such methods, however, lack the accuracy of the former in that the operator cannot so conveniently divide the faceplate, and the backlash in the change wheels, etc., having been released and unaccounted for, may be sufficient to throw the wheel out one or more teeth, thereby producing threads of varying thicknesses. When, therefore, the screw-cutting feed is driven through the turning feed mechanism, the latter should contain a pair of equal wheels—as in Fig. 144—to allow of the sub-division being performed outside the box, where it is accessible and easily seen.

An arrangement whereby the direction of traverse may be reversed from the saddle and the tool returned to the starting point at a much higher rate of speed than that of the feed is shown in Fig. 164.

The device will be more easily understood by considering first the simple reversing train shown in Fig. 163. Here the wheel A is keyed to the spindle or other similar shaft, and drives the wheels B or C, depending on the direction of feed desired. The wheel B is keyed to the change-wheel shaft G, which rotates in the excentric bushes H fitted in the headstock cheeks. The intermediate wheel C is mounted on a shaft carried by the arms, coupled together by a cross shaft and to the bushes H, so that they move together. Thus, if the handle is raised as shown at K, the wheel B meshes directly with A and rotates in the opposite direction; whereas if the handle is depressed as shown at L, the intermediate wheel C is brought into gear with A, thereby causing B to rotate in the same direction as A. By bringing the handle to mid-position, the train is entirely disengaged, as in the ordinary reversing gears. The mechanism can be locked in any of the three positions by a pin in the headstock cheek which enters the excentric bush H, or by any of the other locking devices commonly adopted, and previously described, to suit the position of the train.

There is no particular necessity for the feeding mechanism to be capable of being reversed from the saddle, unless it be to retain the relative position of the tool and work as when screw-cutting. Cross-feeding, or surfacing with a knifing tool, is about the only case where the tool would be employed to cut both ways without having to alter its position. If the proper form of tool is adopted for the traversing feed—be it roughing or finishing—it will drag or score the work if an attempt be made to feed in the opposite direction. When screw-cutting, the thread is generally right- or left-handed—it is extremely rare for it to be both—so that it is impossible by simply reversing the saddle motion to cut both ways, since it would be necessary for the work to rotate in the opposite direction, and the tool to be turned upside down

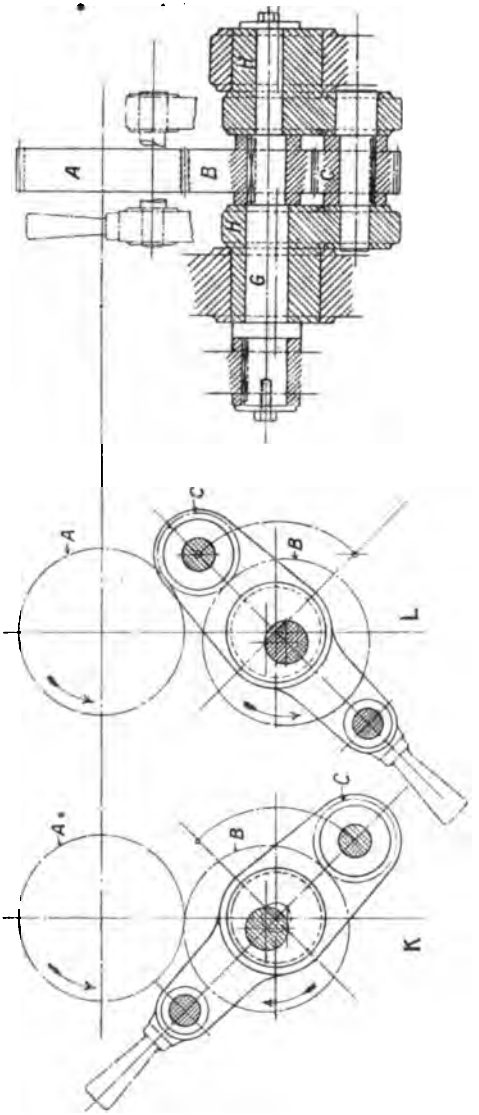


FIG. 163.—SIMPLE REVERSING GEAR.

and probably readjusted. In the device shown in Fig. 164, which is believed to be novel, the wheels A, B, and C form the simple reversing train, as in the arrangement above, whilst D, E, and F form the quick-return train. The wheels B and F are free on the change-wheel shaft G, but can be connected to it by the double-faced clutch I, which is keyed to the shaft. The clutch has a large tooth, so that it can only engage the wheel at one place, and as A and B are equal, the relative position of the clutch and work is maintained. The clutch is controlled from the saddle by a push-and-pull rod at the front of the bed and a lever in the headstock. The wheel D is mounted on the same shaft as the wheel C, and since the latter is smaller than the driver (A or B), and D larger than F, the speed of revolution of the last named is greater than that of the spindle. An intermediate wheel E carried on the right-hand arm H is introduced between D and F, so that whatever may be

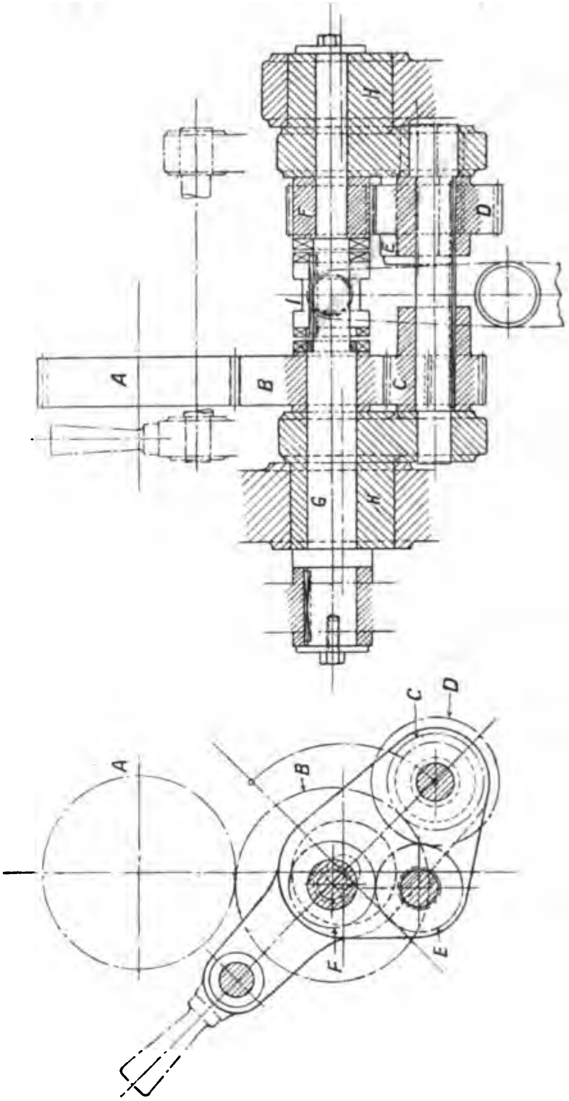


Fig. 164.—Reversing Train.

the direction of motion of B, that of F will be opposite. Should the quick return train not be required, the right-hand train may be disengaged by withdrawing the wheel D.

The accompanying table of change wheels is that often supplied with lathes used for general screw-cutting.

TABLE OF CHANGE WHEELS SHOWING THE PROPER WHEELS FOR CUTTING SCREWS OF VARIOUS PITCHES.

Guide Screw Half-inch Pitch.

Threads in 1 inch.	Wheel on mandril.	Wheel on guide-screw.	Threads in 1 inch.	Wheel on mandril.	Wheel on stud.	Pinion.	Wheel on guide-screw.	Threads in 1 inch.	Wheel on mandril.	Wheel on stud.	Pinion.	Wheel on guide-screw.	Threads in 1 inch.	Wheel on mandril.	Wheel on stud.	Pinion.	Wheel on guide-screw.
1	80	40	7	20			70	30	40	75	15	120	66	20	90	15	110
1¼	80	50	8	20			80	32	40	80	15	120	68	25	85	15	150
1½	80	60	9	80	60	15	90	33	40	90	15	110	70	40	140	15	150
1¾	80	70	10	80	60	15	100	34	40	85	15	120	72	20	90	15	120
2	80	80	11	80	55	15	120	35	40	75	15	140	75	20	75	15	150
2¼	80	90	12	100	75	15	120	36	40	90	15	120	76	25	95	15	150
2½	40	50	13	80	60	15	130	38	30	90	15	95	78	20	90	15	130
2¾	40	55	14	60	70	15	90	39	40	90	15	130	80	30	120	15	150
3	40	60	15	80	90	15	100	40	30	90	15	100	84	20	90	15	140
3¼	40	65	16	60	80	15	90	42	40	90	15	140	85	20	85	15	150
3½	40	70	17	60	85	15	90	44	30	90	15	110	88	20	110	15	120
3¾	40	75	18	50	75	15	90	45	40	90	15	150	90	20	90	15	150
4	40	80	19	50	75	15	95	48	30	90	15	120	95	20	95	15	150
4¼	40	85	20	60	90	15	100	50	40	100	15	150	96	25	120	15	150
4½	40	90	21	80	90	15	140	52	40	120	15	130	100	20	100	15	150
4¾	40	95	22	60	90	15	110	55	40	110	15	150	104	25	130	15	150
5	20	50	24	40	80	15	90	56	50	140	15	150	110	20	110	15	150
5½	20	55	25	40	75	15	100	60	40	120	15	150	112	25	140	15	150
6	20	60	26	60	90	15	130	64	25	80	15	150	120	20	120	15	150
6½	20	65	28	40	70	15	120	65	20	65	15	150	130	20	130	15	150
													140	20	140	15	150

CHAPTER XXXIII

SADDLE-FEEDING MECHANISM FOR HEAVY LATHES.

WHEN the height of centres exceeds 15 in., the saddle is usually fitted with duplex rests to permit of one or more tools on both the back and front rests acting together; the resultant thrust on the job, and the twisting effect on the saddle, being thereby largely obviated. In such cases the feeding mechanism to the saddle is similar to those already described and shown, as, *e.g.*, in Fig. 165. When, however, the centres exceed 24 in., the beds are usually longer, and fitted with two or more saddles, which may or may not have duplex rests, depending on the style of the lathe or class of work for which the machine is intended. If the lathe is to be used for heavy forging and propeller-shaft work, the bed is usually of the two-way type, and the saddles are fitted with duplex rests. For crank shafts, etc., the bed is of the three- or four-way pattern, with independent saddles and rests on the front and back ways, to permit of the tools not only operating practically oppositely to each other, as in the case of the duplex rests, but also upon different parts of the work as may be required. Where the length of the bed is short, or where the initial cost is a prime consideration, the feeding mechanism for lathes with multiple saddles may consist of a change-gear arrangement at the end of the bed and mechanism within the saddle apron similar to that already described for the smaller machines.

In the case of lathes with saddles abreast on three- or four-way beds, a longitudinal feed shaft is usually fitted at the front and back of the bed, and motion is communicated from the spindle and feed-change gear to one of the shafts, and from thence to the other, in which case the saddles all move at the same rate. By providing a separate change-gear device to each shaft, the back carriages may be moved at different rates from those at the front.

Generally, however, in the more recent machines with multiple saddles and long beds, whether of the two-, three- or four-way type, each carriage is provided with independent feeding mechanism, so that it can be actuated in either direction and at varying rates of feed. The feed gear in this instance is placed within the saddle apron after the manner of those described, and driven through change wheels, which, together with a quadrant, are fitted to each saddle, and receive their motion from the longitudinal feed shaft at the back or front of the bed. This last shaft is driven from the spindle, or from a side or faceplate pinion-shaft, by a simple train of gear at the end of the fast headstock. When adapted for screw-cutting, the screw is driven from the spindle through change wheels mounted on a swing frame at the end of the bed, and a single or double clasp nut is fitted to the saddles, as in the smaller machines.

In these large machines the saddles and rests are too heavy to be moved quickly by hand, as, when geared down to allow of hand adjustment, the motion is very slow. The hand-traverse feed is therefore, in such cases, frequently supplemented by a quick-power traversing motion. This is obtained by disengaging the worm and wheel within the saddle apron, and engaging a special set of wheels with the wheel on the rack-pinion shaft, or in some similar way. Another method is to provide a train of gears which can be readily engaged at the end of the feed shaft, or guide screw, whereby the speed of these members is largely increased.

SADDLE MECHANISM FOR LATHE WITH DUPLEX RESTS

In Figs. 166 and 167 is shown a design of saddle fitted with twin-rotating and non-rotating guide screws—one on each side of the bed—for a 36-in. centre surfacing, screw-cutting, and taper-turning lathe, by Hulse and Co., Limited, of Manchester. Each carriage is fitted with a similar mechanism, it being arranged that one may be sliding or screw-cutting whilst the others are surfacing or taper-turning. Motion is transmitted from the spindle through change wheels mounted on the swinging frame at the end of the headstock to the longitudinal feed shaft between the bed ways. A spur wheel—twenty teeth 1-in. pitch—keyed to this shaft, and carried by the bearing within the saddle, meshes with a similar wheel on the short central mitre wheel shaft. Steel mitre wheels keyed to the free sleeves on the transverse hand-adjusting shaft, and capable of being coupled to it by means of a toothed clutch, transmit the motion to the spur wheels—twenty-eight teeth 1-in. pitch—on the rotary nuts on each side of the saddle through a second mitre wheel and pinion, thus furnishing a traversing or screw-cutting motion.

The surfacing and taper-turning motion is obtained through the spur wheel—twenty-two teeth ⅞-in. pitch—keyed to the sleeve of the mitre wheel on the hand-adjusting shaft, and through a similar wheel on the transverse mitre-wheel shaft on the right-hand side of the saddle. The motion is then conveyed to the surfacing screws through the reversing mitres on the swinging brackets at the rear of the saddle, the mitre or worm-change gear free on the end of stud fixed at the rear of the saddle, the spur wheel on same and left-hand transverse shaft to the spur wheels on the end of the surfacing screws at the front of the saddle.

These spur wheels—twenty teeth, four per inch pitch—are all freely mounted on their shafts and engaged to the same by friction cones locked by spanner and nut. Change worm gear is provided to give a varying rate of surfacing or tapering feed; and the independent reversing mechanism at the rear of the saddle to these permits of a reverse taper motion, irrespective of the direction of the traversing feed. Both feeds can be immediately disengaged by the combined handle and nut at the front of the saddle.

The advantages of the twin non-rotating screws are:—

That by propelling each carriage simultaneously by screws at opposite sides of the bed, there is no twisting or cross-straining action upon the

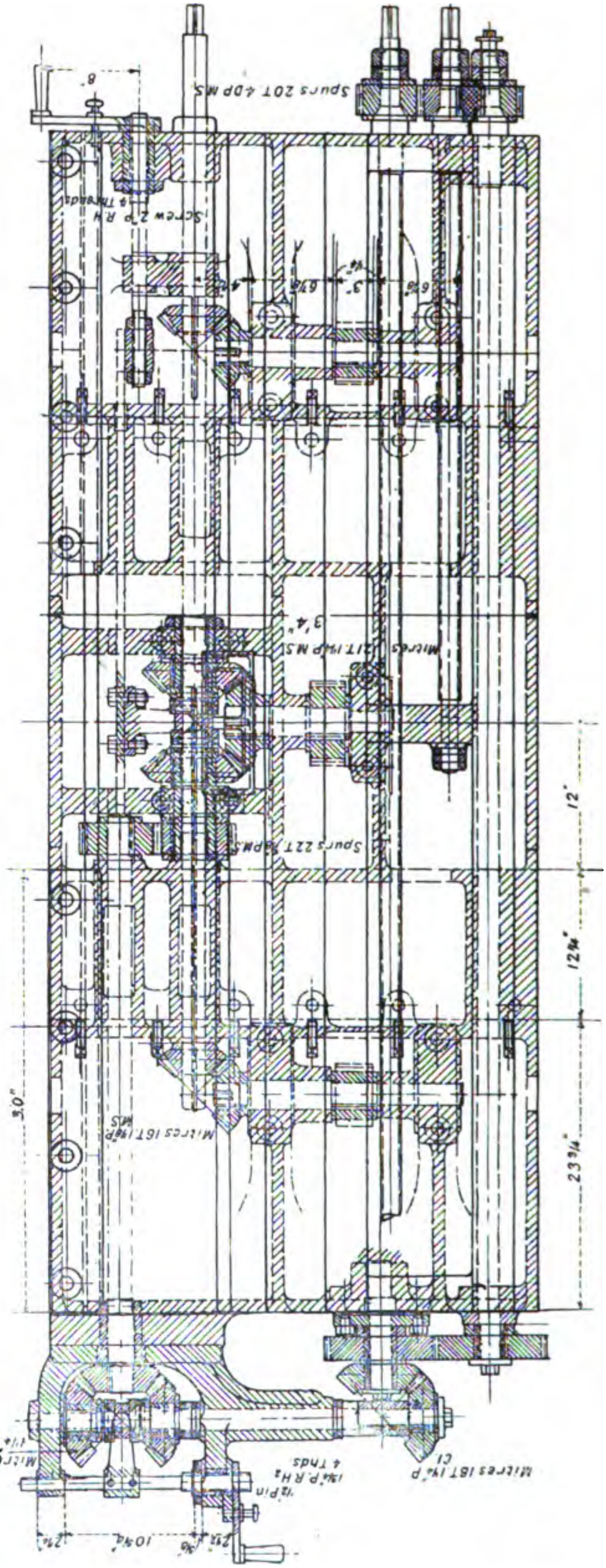


FIG. 166.—LATHE SADDLE—PLAN.

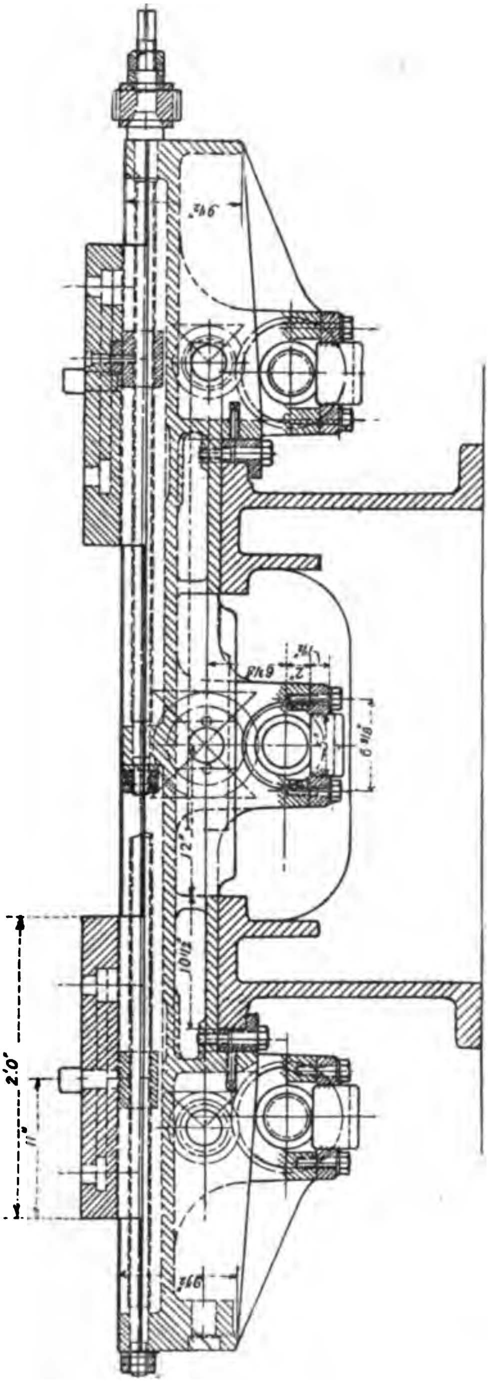


FIG. 166.—LATHE SADDLE—END ELEVATION.

carriages, no matter what the diameter operated upon may be, or whether either or both front and back tools are in operation.

The screws being keyed in their bearings at both ends, the torsional strain is considerably reduced, and a corresponding increased rigidity and cutting power is secured.

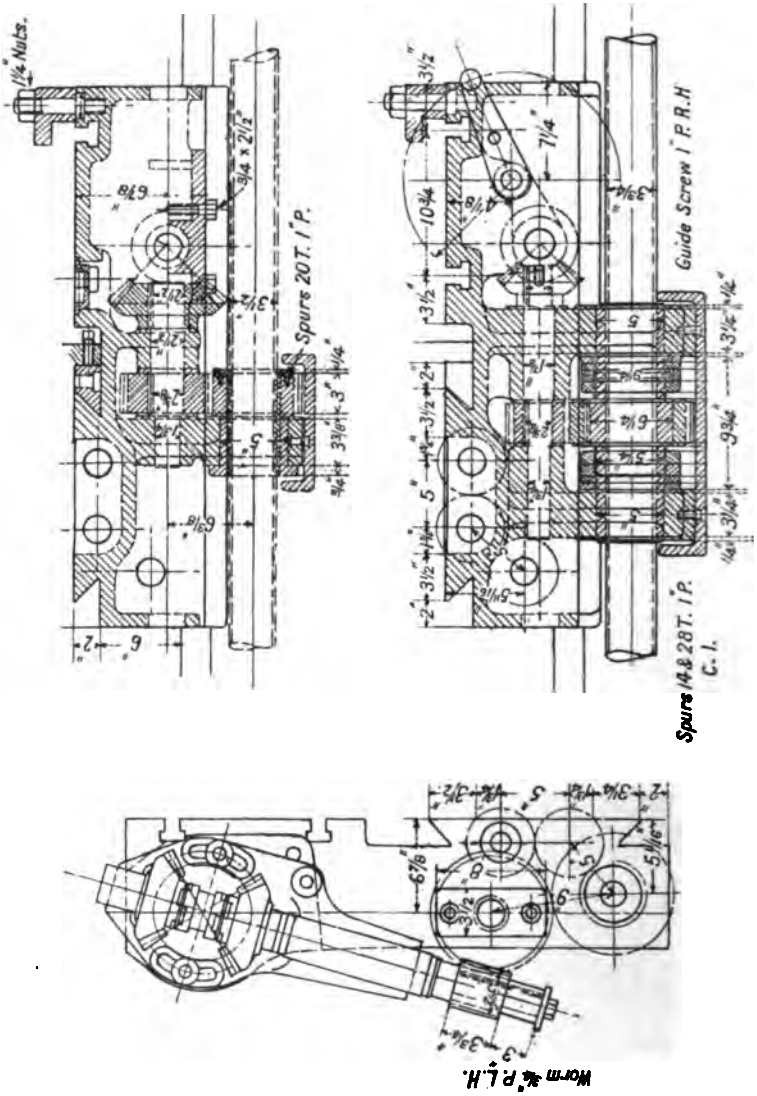


FIG. 167.—LATHE SADDLE—SECTIONAL AND END ELEVATIONS.

SADDLE MECHANISM FOR LATHE, WITH FOUR WAY BED

A general view of a 1500 mm.—60 in.—centre sliding, surfacing, and screw-cutting lathe, by Ernst Schiess, Düsseldorf, is shown in Fig. 168. The lathe is of massive proportions throughout, and will admit 10 metres—33 ft.—between centres. It is intended for turning heavy crank shafts and forgings with high-speed steel. The bed is of the four-way, four-webbed type, 3100 mm.—122 in.—over the ways, and 700 mm.—27½ in.—deep. It

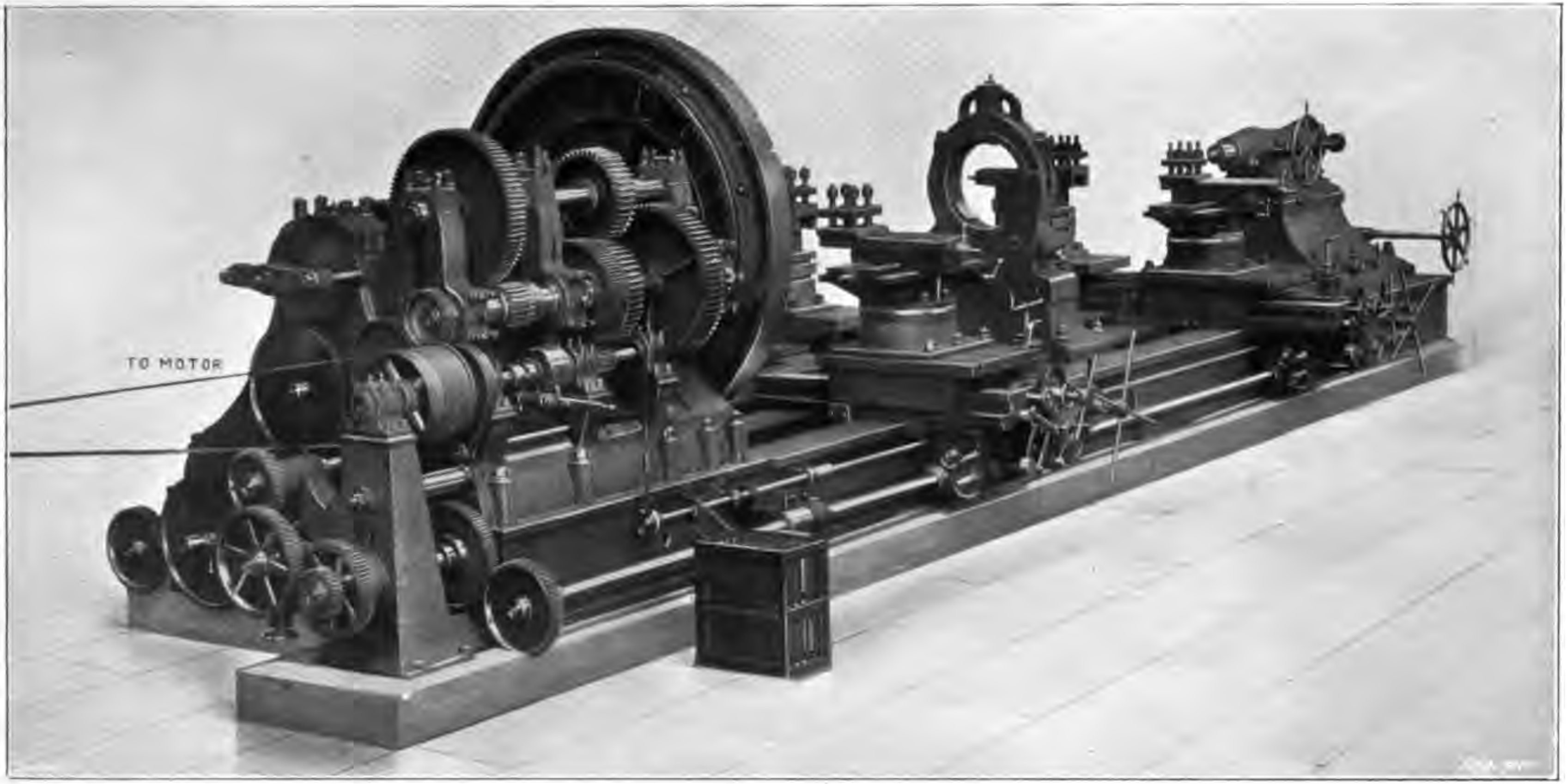


FIG. 168.—60-INCH CENTRE HIGH-SPEED LATHE (SCHIESS).

is fitted with four independent carriages, two on the front ways and two on the back, each being supplied with compound slide rests. The saddles have self-acting, reversible, independent feeds in the longitudinal and transverse directions, actuated by means of a splined shaft at the front and back of the bed, together with a longitudinal quick power and hand traverse. A screw-cutting feed is provided to the front carriages only. The guide screw is placed within the bed and under the front way, and is operated by the change wheels at the end of the fast headstock. This is clearly shown in figure. The longitudinal feed shafts at the front and back of the bed receive their motions from the spindle through the simple train of wheels on the end of the fast headstock. The diagrammatic sketch—Fig. 169—gives particulars of the wheels, and shows the arrangement of the mechanism on each saddle, which, by the aid of the feed tables, will be easily understood. The two

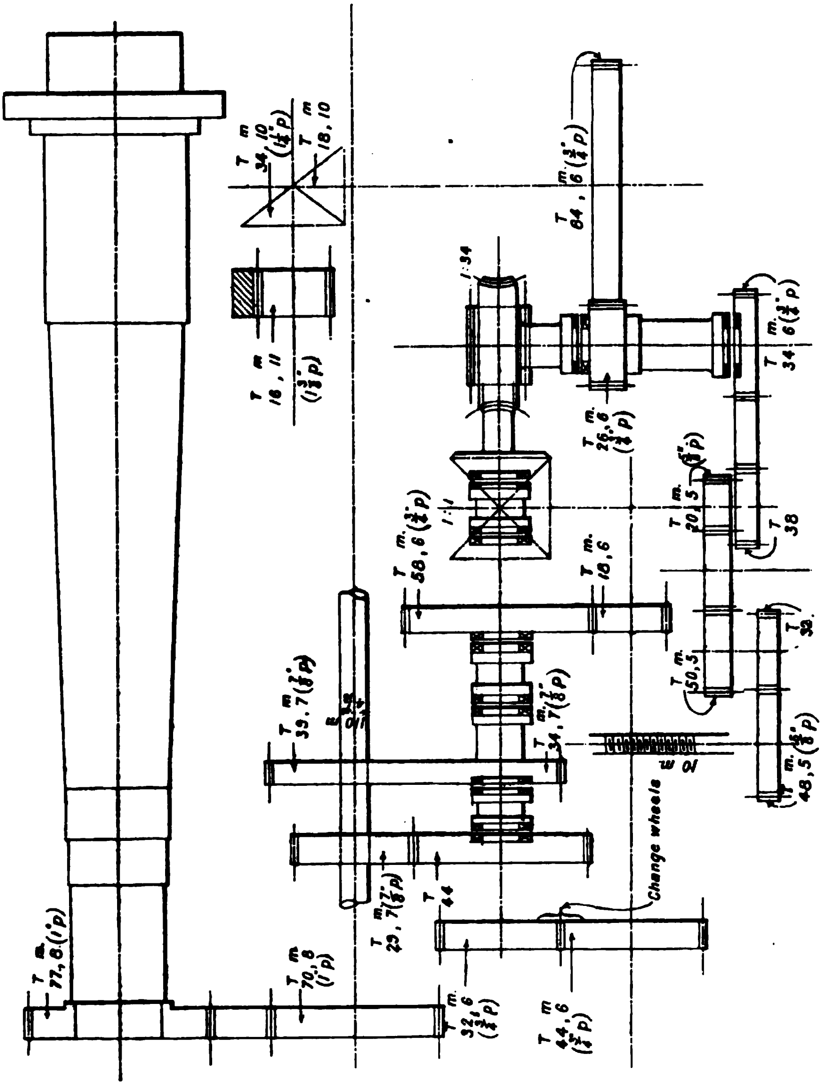


FIG. 169.—ARRANGEMENT OF GEAR IN SADDLE OF SCHIESS LATHE.

wheels on the extreme left of the carriage can be changed to allow of a greater variation of feed if desired.

TURNING FEEDS

Sliding Feed per Revolution of Spindle.

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{1}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 176 \times 3{\cdot}14 = 0{\cdot}57 \text{ mm.} = 0{\cdot}0225 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{39}{34} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{1}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 176 \times 3{\cdot}14 = 0{\cdot}98 \text{ mm.} = 0{\cdot}0386 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{1}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 16 \times 11 \times 3{\cdot}14 = 2{\cdot}5 \text{ mm.} = 0{\cdot}1 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{39}{34} \times \tfrac{1}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 16 \times 11 \times 3{\cdot}14 = 4{\cdot}4 \text{ mm.} = 0{\cdot}175 \text{ in.}$$

Surfacing Feed per Revolution of Spindle.

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{20}{50} \times \tfrac{32}{48} \times 10 = 0{\cdot}435 \text{ mm.} = 0{\cdot}0173 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{39}{34} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{20}{50} \times \tfrac{32}{48} \times 10 = 0{\cdot}76 \text{ mm.} = 0{\cdot}03 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{20}{50} \times \tfrac{32}{48} \times 10 = 1{\cdot}9 \text{ mm.} = 0{\cdot}075 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{39}{34} \times \tfrac{20}{50} \times \tfrac{32}{48} \times 10 = 3{\cdot}35 \text{ mm.} = 0{\cdot}132 \text{ in.}$$

QUICK POWER TRAVERSE TO SADDLES

Advance of Carriage per Revolution of Spindle.

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{38}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 176 \times 3{\cdot}14 = 22 \text{ mm.} = 0{\cdot}87 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{38}{44} \times \tfrac{32}{44} \times \tfrac{18}{58} \times \tfrac{38}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 176 \times 3{\cdot}14 = 38 \text{ mm.} = 1{\cdot}5 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{29}{44} \times \tfrac{38}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 11 \times 16 \times 3{\cdot}14 = 96 \text{ mm.} = 3{\cdot}8 \text{ in.}$$

$$\tfrac{77}{70} \times \tfrac{39}{34} \times \tfrac{38}{34} \times \tfrac{26}{64} \times \tfrac{18}{34} \times 11 \times 16 \times 3{\cdot}14 = 166 \text{ mm.} = 6{\cdot}3 \text{ in.}$$

All the wheels have machine-cut teeth, and the pinions, bevel wheels, and rack are of steel.

Figs. 170, 171, and 172 show a patented design of saddle by Hulse and Co., Limited, of Manchester. The example selected is that for a 32-in. centre-sliding, surfacing, screw-cutting and taper-turning lathe, arranged to admit 58 ft. between the centres, and to swing 46 in. over the saddles. The bed, of the four-girder four-way type, is 69 ft. 3 in. long, 78 in. over the ways, 21 in. deep, and in three lengths. Two non-rotating steel guide screws all of the same length are fitted between each pair of ways. Four independent saddles fitted with compound slide-rests are mounted on the bed—two on the front pair of ways and two on the back. Each carriage is provided with a rotating nut and reversing mechanism. A quick-power traversing motion is furnished for each carriage, and the saddles on the front ways are fitted with taper-turning mechanism.

Motion is transmitted by the mitre wheels on the end of the spindle through vertical and horizontal shafts at the rear of the headstock to the longitudinal feed shafts at the front and back of the bed. The mechanism for the four carriages is alike—with the exception of those on the back ways, which have no tapering motion—and are fitted right and left hand. The diagrams—Figs. 171 and 172—show a left-hand front carriage, the one nearest the fast headstock. On the longitudinal feed shaft within the saddle apron is keyed a sliding clutch, which engages either of the clutches fitted to a pair of mitre wheels. The wheels are free on the shaft, and rotate within bearings in the saddle wings. They mesh with a third mitre wheel keyed to the change-wheel shaft, whose axis is at right angles to the feed shaft. A change-wheel plate swings about the axis of the mitre or change-wheel shaft, and is fitted with suitable wheels communicating with the change wheel on the end of the intermediate wheel shaft. The intermediate wheel—18 teeth, 3 per inch pitch—within the apron meshes with the spur wheels—36 teeth, 3 per inch —freely mounted on the surfacing screw and the transverse feed shaft, but can be connected to the same by the clutches keyed thereto.

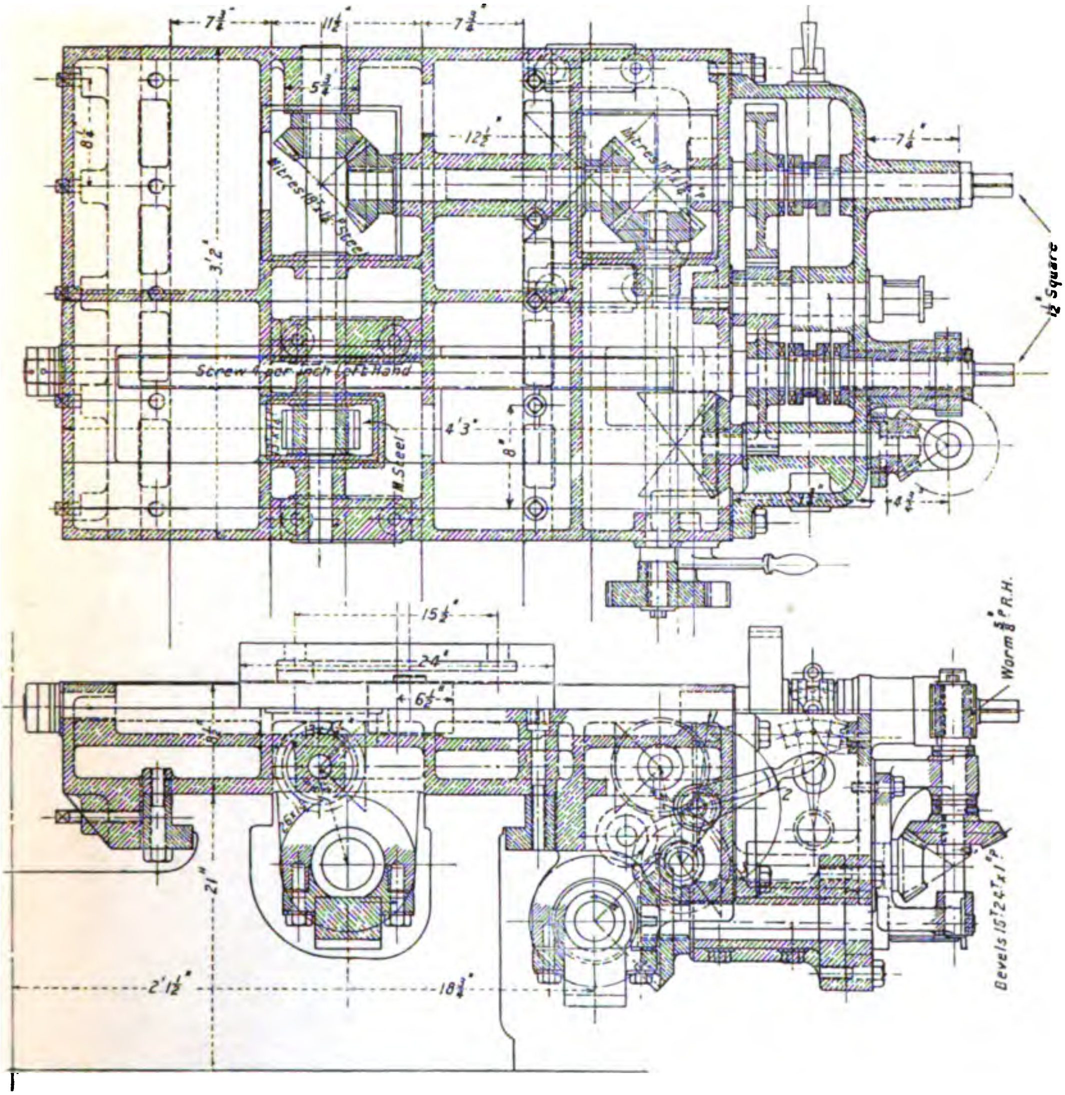


FIG. 171.—SECTIONAL PLAN AND ELEVATION OF HULSE'S SADDLE.

For the traversing and screw-cutting feed, motion is communicated through the mitres on the transverse feed shaft to the mild steel pinion—13 teeth, 1¼-in. pitch—on a short longitudinal shaft to the wheel—20 teeth—keyed to the gun-metal rotary nut within the carriage cheeks. Washers and nuts are fitted to take the thrust and wear in both directions. The screw does not rotate, and is held fast at both ends in its bearings. A right- or left-hand feed is obtained by engaging the clutch on the longitudinal feed shaft with the right- or left-hand mitre. The clutch claw is fixed to a horizontal shaft, screwed at one end, and keyed in the bearing at the other, yet free to slide therein. It is controlled by the combined nut and handle at the right-hand side of the saddle. This reversing mechanism also serves for the surfacing feed. In the

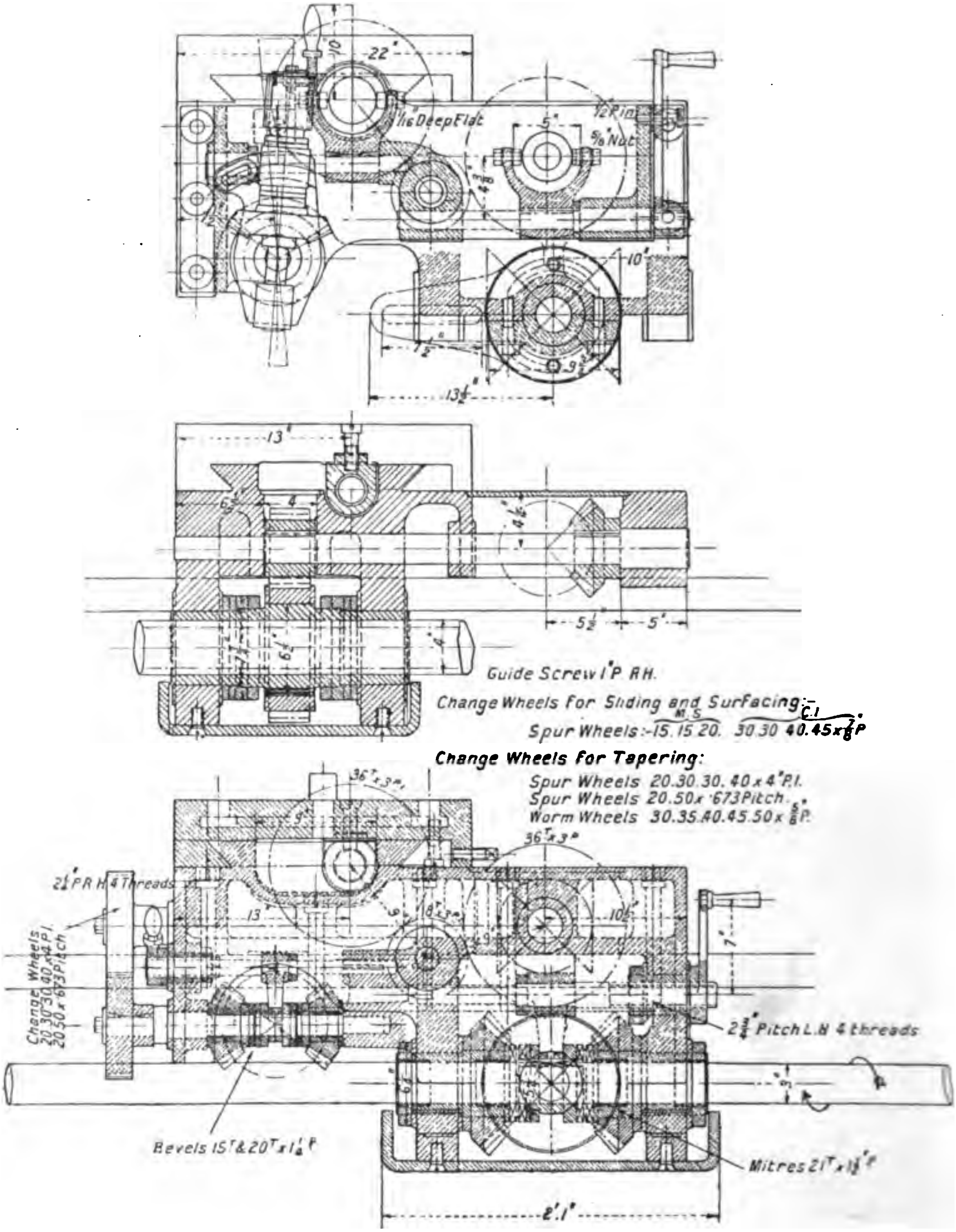


Fig. 172.—Cross-section and Details of Hulse's 32-in. Saddle.

back saddles—which have no tapering motion—the clutches on the surfacing screw and transverse feed shaft are controlled by the one handle, as the shafts which carry the claws are connected by levers. This ensures, by one movement, the entire disengagement of one feed prior to the engagement of the other.

The tapering motion is obtained by combining the traversing and surfacing motions. The mitre wheel on the end of the transverse feed shaft gears with another, keyed to the short lateral shaft near the front of the carriage. On the end of this shaft is mounted a change wheel, which meshes with a similar wheel on the bevel-pinion shaft below—as shown on the drawing—at the left-hand side of the saddle. Motion is transmitted through the bevel wheels to the worm on the swivelling bracket, which meshes with the worm wheel keyed to the loose sleeve on the surfacing screw.

Clutch teeth are formed on the inner end of this sleeve which are engaged with the screw by the same clutch as is used for the surfacing feed proper. Reverse tapers can be cut by engaging the clutch—which is controlled by the handle on the left-hand side of the saddle and mechanism similar to that used for the sliding feeds—with either of the bevel pinions. The worm bracket is made to swivel, to allow of different worm wheels being mounted on the sleeve. As the saddles are provided with their own change wheels and reversing gears, they can screw-cut or feed in either direction irrespectively of each other. Since the change gear may be used in combination with that for tapering, one saddle may be sliding parallel or taper at any required angle and rate of feed, whilst the other is screw-cutting or surfacing.

A list of feeds and tapers, together with the necessary wheels is given in the accompanying table:—

TABLE OF CHANGE WHEELS FOR 32-IN. CENTRE PATENT DUPLEX LATHE.

Change Wheels for Sliding and Surfacing.

Cuts per inch.		Driving wheel on swing frame shaft.	Driven wheels on stud.	Driving wheel on stud.	Driven wheel on intermediate wheel shaft.
Sliding.	Surfacing.				
1	2	45	15	40	30
1½	3	40	15	30	30
2	4	30	20	40	30
3	6	40	30		30
4	8	30	40		30
6	12	20	40		30
8	16	15	40		30
12	24	15	40		45
16	32	30	40	15	45
24	48	15	40	20	45
32	64	15	40	15	45

Change Wheels for Tapering.

Taper required.	Spur wheel on top shaft.	Spur wheel on bottom shaft.	Worm wheel on screw.
1 in 24 in. on diameter	50	20	30
,, 28 in. ,,	50	20	35
,, 30 in. ,,	40	20	30
,, 32 in. ,,	50	20	40
,, 35 in. ,,	40	20	35
,, 36 in. ,,	50	20	45
,, 40 in. ,,	40	20	40
,, 45 in. ,,	40	20	45
,, 50 in. ,,	40	20	50
,, 60 in. ,,	30	30	30
,, 70 in. ,,	30	30	35
,, 80 in. ,,	30	30	40
,, 90 in. ,,	30	30	45
,, 100 in. ,,	30	30	50
,, 120 in. ,,	20	40	30
,, 140 in. ,,	20	40	35
,, 150 in. ,,	20	50	30
,, 160 in. ,,	20	40	40
,, 175 in. ,,	20	50	35
,, 180 in. ,,	20	40	45
,, 200 in. ,,	20	40	50
,, 225 in. ,,	20	50	45
,, 250 in. ,,	20	50	50

QUICK-TRAVERSING MECHANISM TO SADDLES

Fig. 173 shows the quick-traversing mechanism to the saddles of the 32-in. centre lathe of Fig. 170. At the fast headstock end of the bed there is bolted a base-plate, upon which is mounted a 12 horse-power motor. This is coupled directly to the worm shaft. The worm runs in an oil bath, and meshes with a wheel keyed to the hub of the pulley. The pulley has a support in the bearing, and rotates freely on the longitudinal feed shaft at the front of the bed. This pulley communicates with the similarly mounted pulley on the end of the feed shaft at the back of the bed by means of a cross belt. The inside of the rim of both pulleys is machined to receive the split ring and wedge piece, etc.

The disc carrying the expanding ring, wedge piece, and thrusting arm is keyed to the feed shaft. A hemispherical-headed pin is fitted to the end of the arm, and rests upon the cone-ended sleeve keyed to the shaft, and free to slide on the same. On the opposite end of the sliding sleeve is formed a tooth clutch, which meshes with the corresponding clutch keyed to the mitre wheel. It is free on the feed shaft.

For the ordinary turning feeds the positive clutch is engaged with the mitre wheel, and motion is transmitted from the spindle through the mitre wheel and transverse shaft at the end of the bed. To engage the quick withdrawing mechanism, the clutch is moved to the left, so that the cone thrusts the arm outwards, and forces the wedge up the inclined cheeks of the ring. It is thus caused to expand and to bind with the pulley, thereby imparting the motion of the latter to the shaft. The angle of the wedge is sufficiently steep—about 15 deg.—to allow the ring to regain its normal position when the sleeve is withdrawn. A handle is provided for controlling

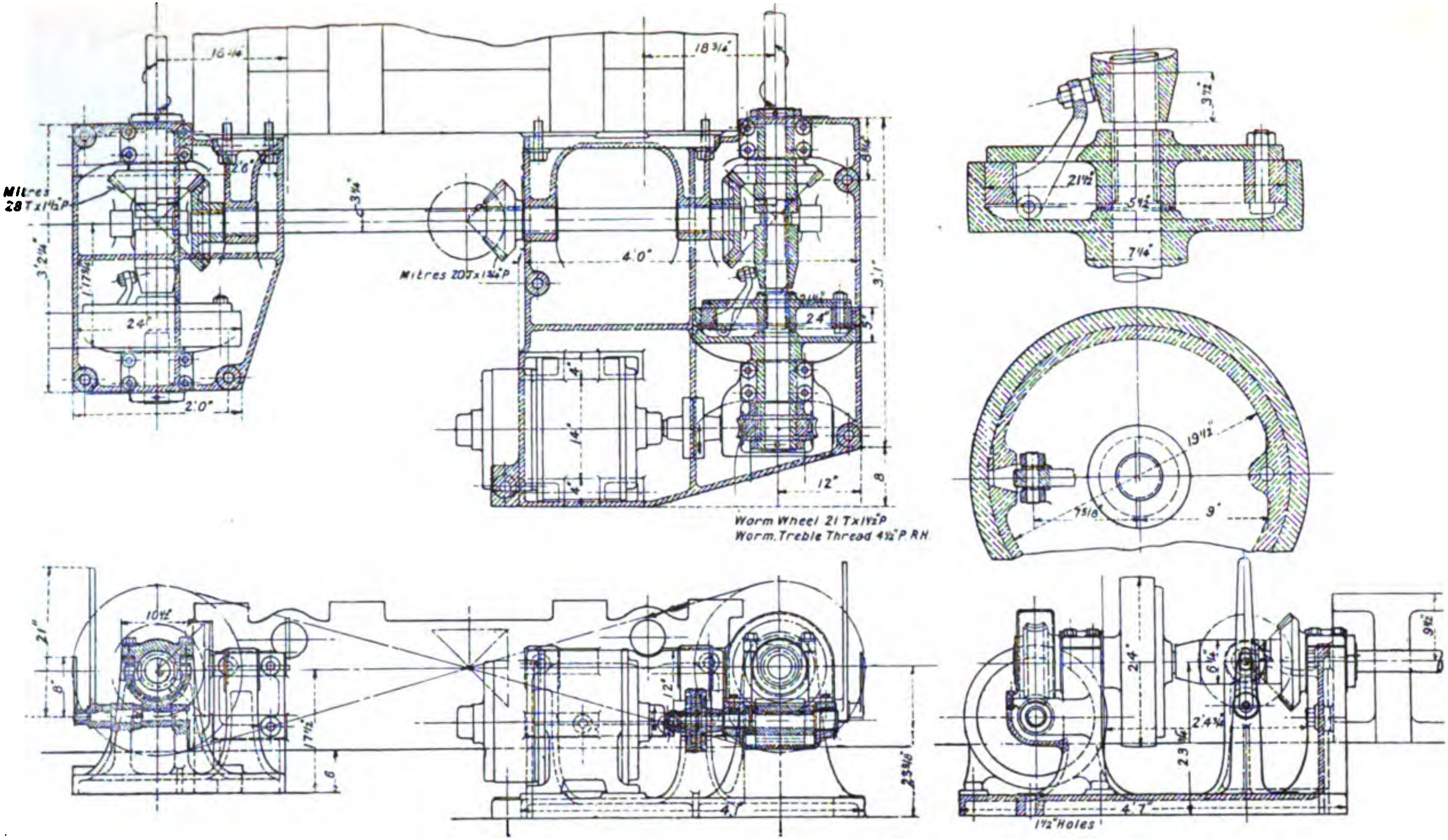


Fig. 173.—Quick-traversing Mechanism, Hulse's 32-in. Lathe.

the sleeve, and another for locking it in any of its three positions. The fact of the shafts being fitted with separate engaging mechanisms permits of either the front or back saddles being rapidly moved along the bed without mutual interference—*i.e.* one or more of the carriages on the back ways may be quickly traversed, while those on the front ways are engaged with the ordinary feeding mechanism, or *vice versâ.* Again, both sets may be moved together, and since the conical sleeve is combined with the positive clutch, it is impossible to have a simultaneous engagement of the quick traversing and feeding mechanism. The motor makes 1000, whilst the longitudinal shafts make 143 revolutions per minute, which gives a traversing speed to the saddles of 3 ft. per minute, when the change wheels for the ¼-in. feed are engaged.

Supports to Guide Screw and Longitudinal Feed Shaft.—It will be observed that in some of the designs discussed a bearing is provided in the saddle for

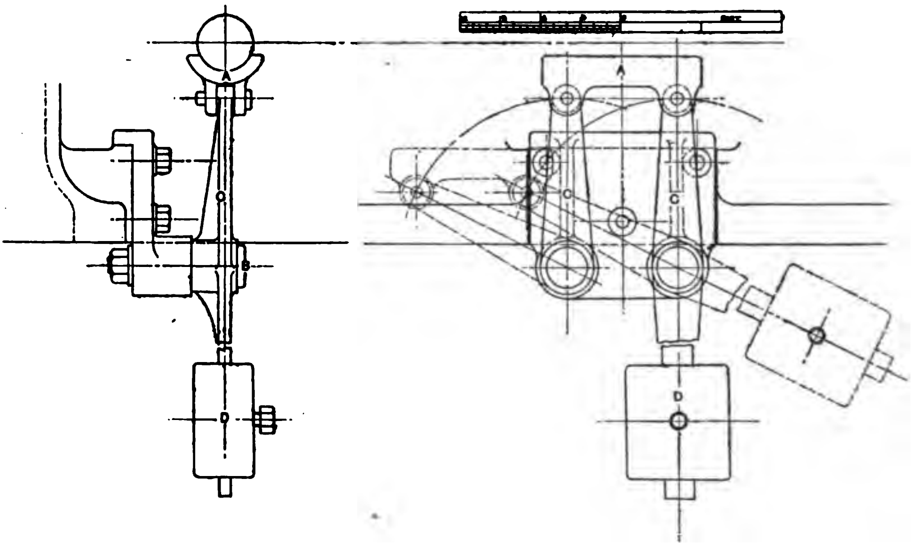


FIG. 174.

the guide screw immediately in front of and behind the nut, and that the longitudinal feed shaft has a bearing on each side of the worm box. The necessity for such supports is apparent if the nut and screw or worm and wheel are to be continually in proper mesh. When the unsupported length of the screw exceeds 36 diameters and the shaft 60 diameters, it is desirable that they should be supported by other means than those mentioned above. As the worm completely surrounds the shaft, it is essential that the bearing should be capable of being depressed to enable the saddle to pass over the same, and also of returning to its first position immediately the saddle is clear. If the nut only embraces a portion of the screw, permanent supports for the same may be fixed to the bed at suitable intervals. When, however, the segmental nut practically encircles the screw so that fixed supports are inadmissible, swing or drop bearings, similar to those employed for supporting the longitudinal shaft, have to be provided.

A swing bearing much used for this purpose takes the form of a bellcrank lever with half-bearings at the extremity of each arm, and cut away at the fore part to clear the shaft when swinging. The design is not suitable for lathes with multiple saddles, and the scheme shown in Fig. 184 is generally preferred. The upper portion or bearing A is carried by the two levers C, swinging about the fulcrum pins B fixed in the bracket attached to the bed. The bearing is counterbalanced by the weight D fixed to the extremity of one or both levers. A striking plate fitted to the saddle, and coming in contact with A or other provided projection, depresses the bearing to the position shown by the dot-and-dash lines. Although this type of bearing is very commonly used, it is not as satisfactory as it would appear. When the bearing is clear of the saddle, the balance weight brings it quickly back, thereby causing it to surge and, more often than not, will come to rest before it has regained its central position. The weight of the shaft coming upon it throws it further out of line, and to increase the balance weight so as to lift the shaft aggravates the surging effect. Even when most accurately fitted and balanced, the unstability of the design and the vibration of the shaft and machine would prevent it maintaining its proper position should it happen to regain it.

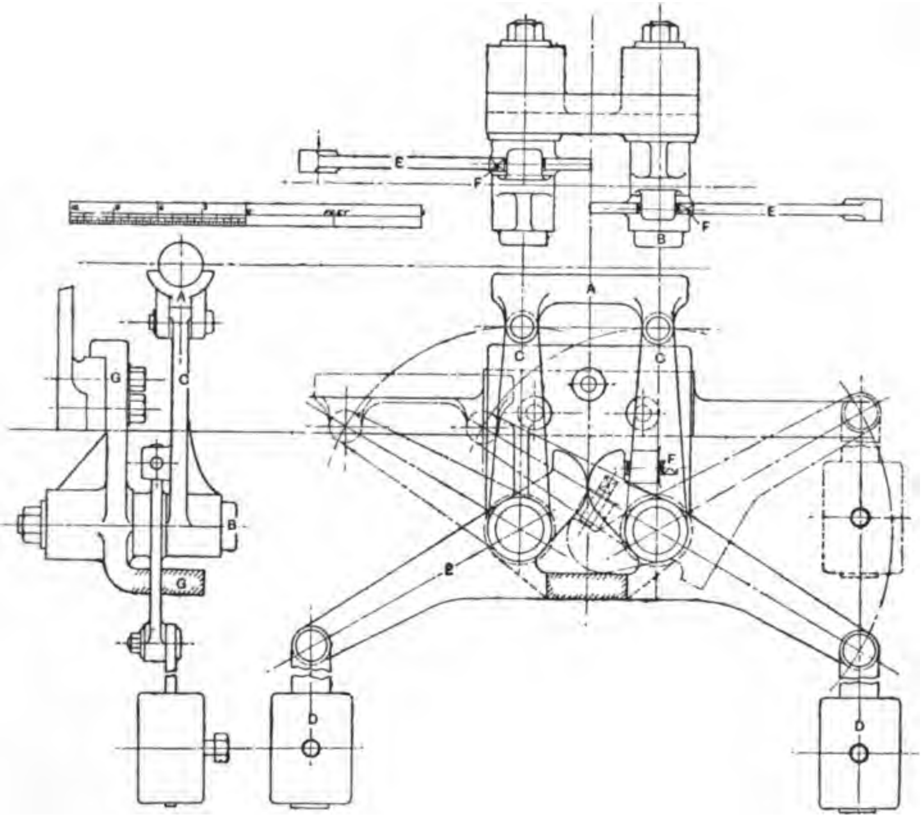


FIG. 175.

In the design by Hulse and Co., shown in Fig. 175, the bearing A is carried by the two single levers C, and independent levers E are employed to carry the balance weights D and their bars. Both C and E are free to rock separately on the fulcrum pins B, fitted in the bracket G bolted to the bed. This bracket (G) is furnished with a toe-piece (etched lined in figure), against which the heel of the levers E is made to bear when the bearing is in its normal position, whilst the toe of the levers bear against the adjustable screws F fitted in the lugs of the lever C. Thus the levers E and weights D are alternately raised and freed from G as the bearing is moved in opposite directions. If the saddle is moving towards the left hand or fast headstock, the bearing is pushed in that direction and downwards, and the screw F in the right-hand lever C simultaneously thrusts against the adjacent lever E, and raises the balance weight as shown in dotted lines. Meanwhile the left-hand lever E is held in its normal position by the toe G. Conversely, when the bearing is moved to the right hand, the left-hand lever and weight is raised whilst the right-hand members are retained. From this it will be observed that no matter in what direction the bearing may be moved, it is brought back to its proper position and against a positive stop so that it cannot surge. The counterweight acting at an advantage is capable of preventing the bearing being disturbed by the vibration of the machine, etc., and the shaft from sagging.

CHAPTER XXXIV

STRENGTH AND PROPORTIONS OF FEED MECHANISMS

FUNDAMENTAL to the rational design of the feed mechanism is a knowledge of the force required to slide the saddle along the bed whilst taking a cut. This force depends on: (1) The magnitude and direction of the resultant cutting pressure on the tool; (2) the design and proportions of the saddle and bed, together with the relative positions of the feeding rack or feed screw and tool point; (3) the coefficients of friction between the various sliding surfaces of the saddle and sheers.

The three rectangular components of the resultant cutting force have values which are known from our experiments with the "Universal lathe tool dynamometer." These experiments have been described and tabulated in Chapters III. and IV.

For soft or medium steel the vertical component V has a value of about 100 tons per square inch of area of cut, when working with a tool ground to ordinary shop angles. The horizontal component T parallel to the length of the bed or "traversing force" has for the same material and shape of tool a magnitude equal to roughly 0·25V; and the horizontal component S perpendicular to the length of the bed, called by us the "surfacing force," may be taken to be equal to 0·5V.

In accordance with these data the total force required to slide the saddle along the bed will consist of:—

(1) The net traversing force $T = 0{\cdot}25V$.

(2) The frictional resistance between the saddle and top of bed, due to the vertical cutting force V, together with the weight of saddle and rests. If the coefficient of friction for such greasy but polished surfaces be taken equal to 0·15, this force of friction will amount to $0{\cdot}15 \times 1{\cdot}1V = 0{\cdot}165V$; since the weight specified is about 10 per cent. of the force due to the standard cut for lathes up to 36-in. centres.

(3) The frictional resistances between the saddle and the edges of the bed are due either to the surfacing force S as a whole; to forces produced at two opposite corners by the "cross-wind" of the saddle, owing to the rack and tool point not being in the same vertical plane; or to the backward tilt caused by the height above the bed at which the force S acts. With $\mu = 0{\cdot}15$, as before, we have—since $S = 0{\cdot}5V$—for this frictional resistance, $\mu S = 0{\cdot}15 \times 0{\cdot}5V = 0{\cdot}075V$. It is easy to show that the addition to be made to this for the friction due to the cross-winding and tilting tendencies will not ordinarily amount to more than 0·01 to 0·02V. Thus the total

frictional resistance due to the action of the surfacing force S amounts to about 0·1V.

The net force F required to slide the saddle along the bed at a uniform speed, constituted by the sum of these three items is, therefore:—

$$\begin{aligned} F &= (0{\cdot}25 + 0{\cdot}165 + 0{\cdot}1)V \\ &= 0{\cdot}515V, \text{ or, say } \tfrac{1}{2}V \end{aligned}$$

With the standard cut, of area $= \dfrac{h^2}{6400}$ square inches, this force has values for the various sizes of lathes given by the following expression [in terms of the height of centres (h)]:—

$$\begin{aligned} F = \tfrac{1}{2}V = \tfrac{1}{2} \times 100 \times 2240 \times \frac{h^2}{6400} &= 17{\cdot}5h^2 \text{ lb.} \\ &= 18h^2 \text{ lb.} \end{aligned}$$

Thus with—	$h = 6$ in.	12 in.	18 in.	36 in.	72 in.
we have—	$F = 648$	2590	5830	23,400	93,600 lb.

The power-drive for supplying this pushing force may be obtained in very many different ways, as described in Chapters XXXI. and XXXIII.; but that force is delivered to the saddle in the end by means of either a pinion hung from the saddle, operating upon a rack fixed to the bed, or by a nut on the saddle driven from—or driven by—a screw alongside the bed.

The teeth of this pinion, which is usually of steel and machine cut, have therefore to be of sufficient strength and rigidity to transmit the force $F = 18h^2$. On the assumption that the pinion should not have less than fourteen teeth, that these teeth are to have radial flanks, that their height is to be 0·55 times their pitch, and that the pinions are to be of a width equal to three times the pitch; then, in the fundamental formula for the strength of a machine-cut tooth (Chapter XXIII.)

$$F = P = \frac{ea^2}{6\beta}pbf \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (135)$$

we shall find that $e = 2$, $a = 0{\cdot}434$, $\beta = 0{\cdot}55$, and $F = \dfrac{2 \times 0{\cdot}434^2}{6 \times 0{\cdot}55}pfb$ $= 0{\cdot}114pbf$.

$$\text{But } b = 3p \quad \therefore F = 0{\cdot}342p^2f = 18h^2. \quad . \quad . \quad . \quad (136)$$

This shows that the pitch should vary directly as the size of the lathe.

According to the diagram—Fig. 176—in which the results obtained from practice are plotted as ordinates on a base of lathe heights, we find, however, that the law is

$$p = \frac{\sqrt{h}}{5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (137)$$

or the pitch increases more slowly than the height of centres.

There are three possible causes for this discrepancy between theory and practice—

(1) The values adopted in practice for the smaller sizes may be too great

(2) Those used for the larger sizes may be too small; or

(3) If the values adopted in practice are really correct, the safe working stress allowed does not remain constant for all sizes of lathes, but is diminished as the rack pinion gets smaller in diameter. If the stress (f) be taken equal to $1320h$ in the fundamental formula above, we immediately obtain the practical rule

$$p = \frac{\sqrt{h}}{5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (138)$$

A simple formula which almost equally well fits the data of practice, and provides a greater margin of strength for the smaller sizes is

$$p = \frac{h}{60} + \tfrac{1}{2}\text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (139)$$

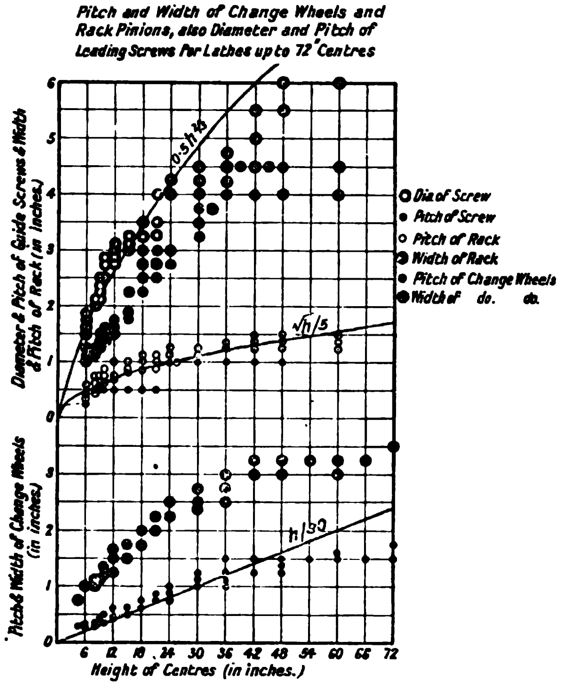


FIG. 176.

Feed Shaft.—In proportioning the feed shaft, it is proposed to make it of such strength and stiffness as to be able to transmit the power necessary for traversing the saddle at the standard rate of feed $\left(\frac{h}{160}\right)$ with the standard area of cut $\left(\frac{h}{40} \times \frac{h}{160}\right)$ when it is itself rotating at the same speed as the spindle. The reason for this will be clearly understood from the annexed skeleton sketch—Fig. 177—of an ordinary feed mechanism, in which four change gears are shown corresponding with our proposed four standard rates of feed, $\frac{h}{108}$, $\frac{h}{160}$, $\frac{h}{240}$, and $\frac{h}{360}$, and in which for the $\frac{h}{160}$ feed the feed shaft is shown geared right back to the spindle by equal wheels. With a worm having t threads let the rack pinion make n revolutions, whilst the feed shaft makes one turn. Then, K being the number of teeth in the worm wheel, L and M the number of teeth in the wheels so marked on the figure—

$$n = \frac{t}{\mathrm{K}} \cdot \frac{\mathrm{L}}{\mathrm{M}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (140)$$

Now the rack pinion circumference is $14\frac{\sqrt{h}}{5}$, and the saddle therefore moves $14\frac{\sqrt{h}}{5}n$

$$= 2{\cdot}8n\sqrt{h}$$
$$= \frac{14t\mathrm{L}}{5\mathrm{KM}}\sqrt{h}$$

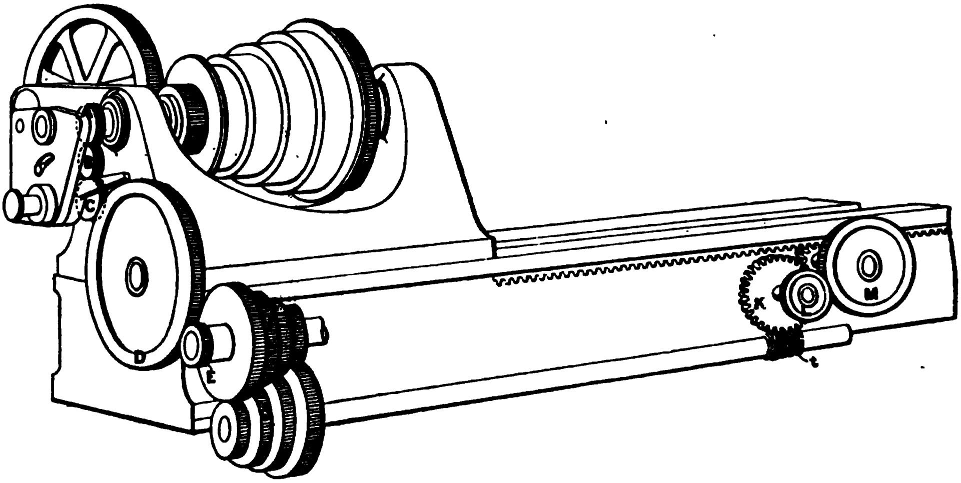


FIG. 177.

whilst the feed shaft makes one turn. If this distance $= \frac{h}{160}$, we must have —taking $\frac{\mathrm{L}}{\mathrm{M}} = \frac{1}{4}$—

$$\frac{h}{160} = \frac{14t}{5 \times 4\mathrm{K}}\sqrt{h}$$

or

$$\frac{\mathrm{K}}{t} = \frac{112}{\sqrt{h}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (141)$$

Thus if $\frac{\mathrm{L}}{\mathrm{M}}$ be taken at ¼ and the ratios given in the table are adopted for the worm gear ratio $\left(\frac{\mathrm{K}}{t}\right)$, the feed shaft will rotate once whilst the saddle is fed through a distance $= \frac{h}{160}$ inches (*i.e.* whilst the spindle rotates once).

Table of worm gear ratios on feed shaft $\frac{\mathrm{K}}{t}$.

Thus for $h =$	6 ...	12 ...	18 ...	36 ...	72
$\frac{K}{t}$	46 ...	32 ...	26 ...	19 ...	13

Let the force acting on the teeth of the wheel G be Q lb. and its pitch circle radius be r inches; then the work done on G during one revolution of the feed shaft is $2\pi Qr$ inch-lbs. During the same interval of time the work spent in driving the saddle (through $\frac{h}{160}$ inches) is $0{\cdot}5V \times \frac{h}{160}$ inch-lbs. If, therefore, the efficiency of the drive as between G and the saddle be $0{\cdot}25 \times 0{\cdot}94 \times 0{\cdot}94 = 0{\cdot}22$ (0·25 for the worm drive, 0·94 for each of the pinion drives), we shall have

$$2\pi Qr \times \eta = 0{\cdot}5V \times \frac{h}{160} = 18\frac{h^3}{160}$$

and

$$Qr = \frac{18h^3}{160 \times 2\pi\eta} = \frac{h^3}{12{\cdot}3}\text{ lbs. inches.}$$

or

$$T = \frac{h^3}{12}\text{ lbs. inches} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (142)$$

is the torque to which the feed shaft is subjected.

Now it may be easily proved that if $\frac{\theta}{l}$ be the angle of twist allowed per unit length of the shaft, d its diameter, and G the modulus of rigidity of its material, then

$$\frac{\theta}{l} = \frac{10{\cdot}2T}{Gd^4}$$

Hence

$$d^4 = \frac{10{\cdot}2T}{G\frac{\theta}{l}} = \frac{10{\cdot}2h^3}{12G\frac{\theta}{l}}$$

or

$$d^4 = ch^3 \quad \text{or } d = \frac{h^{\frac{3}{4}}}{c}$$

when c is a constant depending on the rigidity of the material of the shaft, on the angle of twist per unit of length allowed in practice, and on the depth of the keyway in the shaft. From the data obtained in practice it has been found that $c = 4\frac{1}{2}$ gives a fair agreement with the sizes ordinarily employed. So that the feed shaft diameter may be found from the formula—

$$d = \frac{h^{\frac{3}{4}}}{4\frac{1}{2}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (143)$$

Feed shaft diameters.

$h =$ 6	12	18	24	30	36	48	60
$d =$ 1 in.	1½ in.	2 in.	2½ in.	2⅞ in.	3¼ in.	4 in.	4⅞ in.

Strength of Teeth for Feed-change Gear.—The pitch of the feed-change gear teeth is found to be about $\frac{h}{30}$—*vide* Fig. 176. The number of teeth on the wheel G is about 30; so that the diameter of G is $\frac{h}{30} \times \frac{30}{\pi} = \frac{h}{\pi}$. Hence, the force Q acting on the teeth is known from $Q = \frac{h^3}{12r} = \frac{h^3 \times 2\pi}{12h} = \frac{h^2\pi}{6} = \frac{1}{2}h^2\sim$.

Taking, as before, in the formula $\frac{ea^2}{6\beta}pbf$; $e = 2$, $a = 0{\cdot}434$, $\beta = 0{\cdot}55$, and $b = 2p$ we get—

$$\text{the load} = 0{\cdot}228\,p^2f = Q = \tfrac{1}{2}h^2.$$

Hence, $$p^2 = \frac{h^2}{0{\cdot}456f};\ i.e.\ p = \frac{h}{0{\cdot}675\sqrt{f}}$$

in which if f be taken equal to 2000 lb. per sq. in., we obtain the practical formula—

$$p = \frac{h}{30} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (144)$$

Comparing this rule $p = \frac{h}{30}$ with the data plotted in the diagram, we see that for small sizes the teeth are usually of greater, and for large sizes, of smaller pitch than the rule indicates.

This appears to show, as was found to be the case with the rack pinion, that a smaller working stress is usually allowed for small than for medium sized teeth.

Another reason for the discrepancy, with lathes of larger sizes than 36-in. centres, may be that for them the feed shaft is often run at a higher speed than the spindle, not only when the lathe is taking the standard cut with the standard feed, but for all cuts and feeds. The change gear is situated in the saddle itself, as pointed out in a previous article, and the feed shaft to spindle velocity ratio is constant.

The above rule for the pitch is applicable to all the wheels of the change feed mechanism. They are usually all of the same pitch, although the reversing train on the spindle is generally of steel. This is justified by the fact that the train is in use for every cut and feed.

THE LEADING SCREW

The leading screw is always of superabundant proportions as regards either strength or stiffness, and its design, apart from kinematical considerations, which have been already very fully gone into, appears to depend upon the allowable bearing pressure between nut and thread, *i.e.* upon the amount of wear to be provided against.

The pitch is usually two threads per inch up to and including 21-in. centres, and one per inch for sizes beyond that.

If p is the pitch, d the mean diameter of thread, and m the ratio of length of nut to mean diameter of thread $\left(\frac{l}{d}\right)$, the area of contact with the nut is—

$$\tfrac{1}{2}p\frac{\pi d}{2}\,\frac{md}{p} = \frac{WB\pi}{4}d^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (145)$$

Here we assume that only the upper half of the nut is in use. The normal thrust between nut and screw is about $20h^2$ lb. Hence if q is the safe bearing pressure we must have—

$$q\frac{m\pi}{4}d^2 = 20h^2$$

from which

$$d^2 = \frac{80}{qm\pi}h^2 = \frac{9h^2}{q}; \text{ if } m = 3$$

If q were constant for all sizes of screw and equal to 180 lb. per square inch, then $d^2 = \frac{9}{180}h^2 = \frac{h^2}{20}$ and $d = \frac{h}{4\frac{1}{2}}$.

The law exhibited on diagram 184 from practical experience is

$$d = \frac{h}{5} + \frac{1}{2} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (146)$$

for sizes up to 18-in. centres.

For sizes above this the value allowed for the bearing pressure (q) may be greater.

According to Osborne Reynolds, "Works," vol. ii. p. 265, eqn. 35, the allowable load per unit area varies as the square of the smallest dimension. Hence, for a thread of 1-in. pitch the value of q should be 720 lb. per sq. in. if 180 be correct for a thread of $\frac{1}{2}$-in. pitch.

Hence,

$$d^2 = \frac{9}{q}h^2 = \frac{9}{720}h^2 = \frac{h^2}{80}$$

$$\text{or } d = \frac{h}{9} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (147)$$

The actual values are approximated to by linear law.

$$d = \frac{h}{13\frac{1}{2}} + 2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (148)$$

Showing that either a smaller bearing pressure is allowed as the diameter diminishes, even though the pitch remain the same; or that the proportion of length to diameter of the nut is less as the size increases.

A similar method of calculation to that employed for the stiffness of the feed shaft might be made use of for determining the diameter of the leading screw. The ratio of the force acting tangentically at mean thread diameter to the saddle reaction F is about 1 to 5 for ordinary pitches and diameters. The diameter is then found to increase as $h^{\frac{3}{4}}$, and the formula $d = \frac{1}{2}h^{\frac{3}{4}}$, which is obtained in this way, seems to agree tolerably well—for the smaller sizes, at all events—with the data of practice—see Fig. 176.

CHAPTER XXXV

SADDLES AND RESTS

If the lathe is intended for plain shafting and similar work, the rest may consist of a simple pillar mounted on the saddle to support the tool. This class of work necessitates a movement of the tool in the longitudinal direction for operating on the circumference of the bar, and another in the transverse direction for facing the ends, etc. These motions (or feeds)—longitudinal or traversing, and transverse or surfacing—are generally self-acting; the former nearly always so as the work is usually long in proportion to the diameter, and the latter depending on whether the nature and amount of the work justifies the increased cost of its inclusion. It is necessary to provide hand adjustments in the above directions to permit of the tool being set to the proper cut or feed, and of finishing the work to definite dimensions, etc. The longitudinal hand feed may be obtained by means of a fine hand-traverse mechanism in the saddle apron, as already described, or by a supplementary slide parallel to the bed, and mounted on the saddle. The transverse adjustment is secured by a cross-slide on the saddle, upon which the tool rest is mounted. A simple rest of this description is frequently supplied with crankshaft lathes for machining the crank pins and webs (Fig. 178). Here the

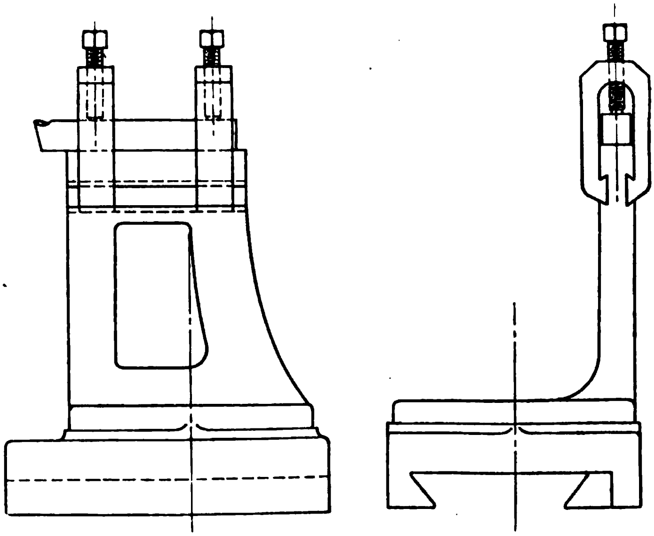


Fig. 178.

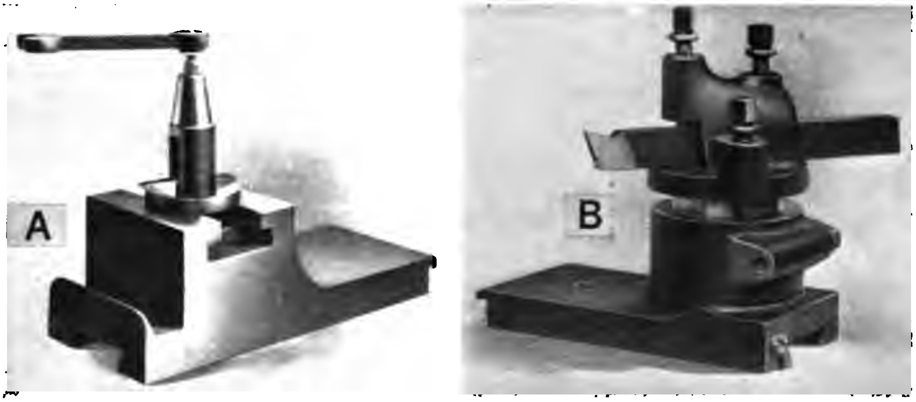


FIG. 179.—AMERICAN RESTS FOR SMALL AND MEDIUM SIZE LATHES.

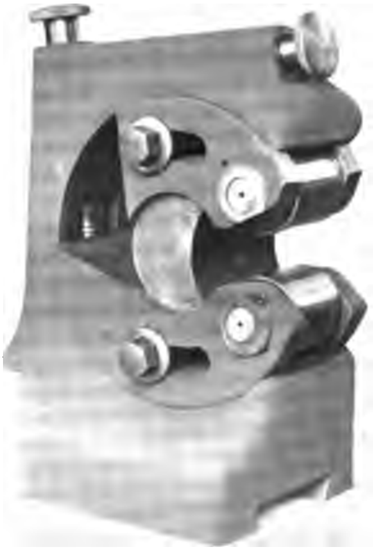


FIG. 191.—ROLLER FOLLOWER REST (LODGE AND SHIPLEY).

compound rests are replaced by a pillar secured to the transverse slide on the crossways of the saddle by bolts fitted in the circular tee slot of the slide. The pillar with the tool is thus free to occupy any position in the horizontal plane. This class of rest is extensively used in America, even on lathes adapted for screw-cutting.

In the American design of simple rest (Fig. 179, A), the transverse slide is provided with a tee slot running parallel with the work, and in this is fitted a circular post having an adjustable ball-faced packing piece to carry the tool. The tool post or holder has a circular base to allow of it swivelling in the tee slot. This type of rest has the advantage of cheapness, being simple and easy to construct; and the increased rigidity due to the small number of joints permits of heavier cutting and a more accurate finish.

In the engineering establishments of this country the work is generally of a very varied character, for which the above description of rest is unsuitable.

In screw-cutting, for example, it is a practical impossibility to cut the full width and finish the thread of a moderately coarse-pitched screw at one setting of the tool. To secure the necessary finish the operator usually takes a cut down each side of the thread in succession, and more frequently than not the tool has to be reground after roughing out the thread. From what has been said in Chapter XXXII. on screw-cutting, it will be apparent that a longitudinal adjustment to the tool cannot be made by the saddle hand-feed mechanism after the thread has once been started. The tool may, of course, be tapped by a hammer or the change wheels shifted, and the carriage advanced the required amount prior to the re-engagement of the wheels, but such methods are too crude and indefinite to be considered.

In order, therefore, to obtain the required adjustment for finishing the thread, after resetting the tool subsequent to grinding, etc., the rests must be fitted with a slide running parallel to the length of the bed or work. This slide is usually made of a fair length, and mounted on a circular base so that it may swivel on the incutting slide, which is provided with an annular tee slot and bolts for securing the slide thereto. It is thus particularly handy for boring either parallel or taper holes, turning up centres, or quick tapers which cannot be done by the former bar; or which, if the number be few and of short length, may be done much more quickly by the slide than the bar.

In addition to the above, many other conveniences are secured (obvious to those who have used a lathe) by the inclusion of the swivel slide in the rests of a general-purpose lathe. The increased handiness far outweighs the greater cost of the same, even in the smaller sizes of machines. In some of the larger lathes a self-acting, as well as a hand-feed, motion is fitted to the swivel slide. In these large machines the weight of the rests is so great as to considerably affect the delicacy of the hand transverse adjustment; and for this reason many are fitted with a supplementary slide immediately below the tool slide, as shown in the 32-in. lathe by Hulse (Fig. 180).

The transverse ways on the saddle should be of such a length that the tool in the top rest can operate on the periphery of the largest diameter swung in the faceplate, and can travel across the face of same to the centre without disturbing the position of the tool.

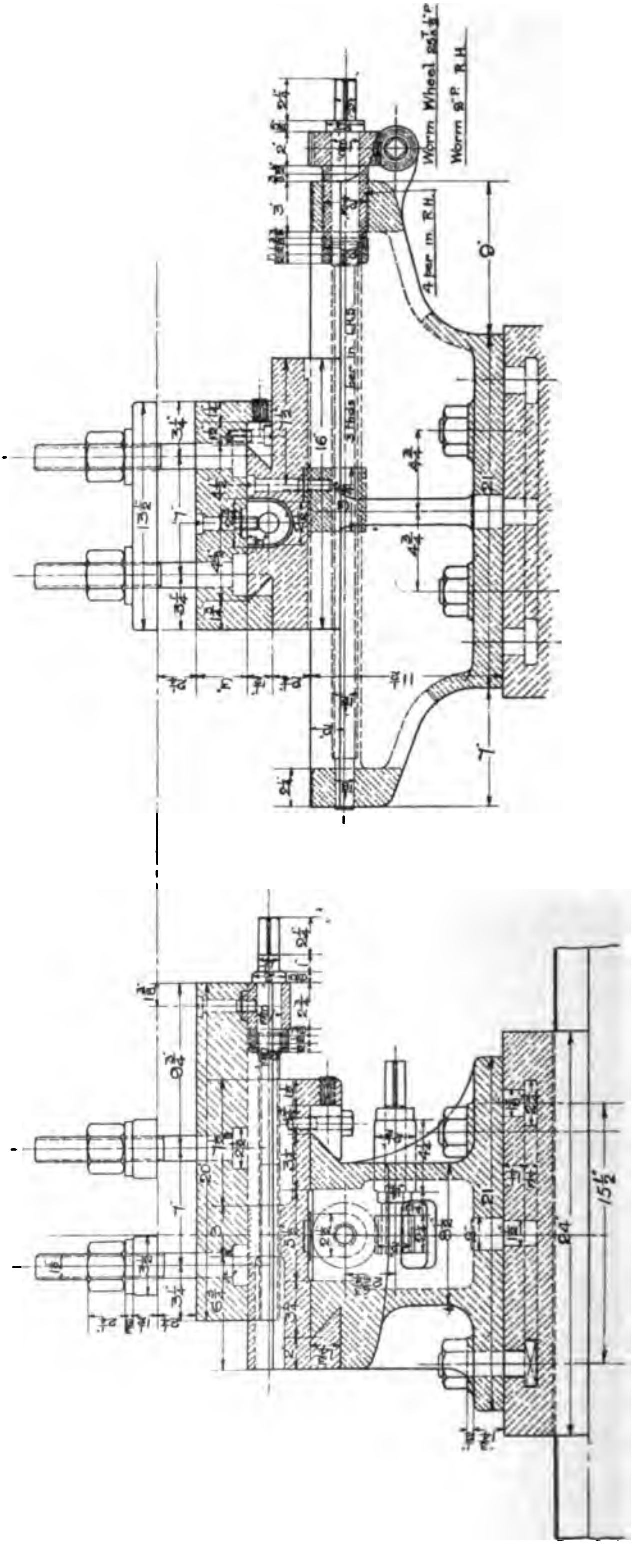


FIG. 180.—FRONT RESTS FOR 32-INCH CENTRE LATHE (HULSE)

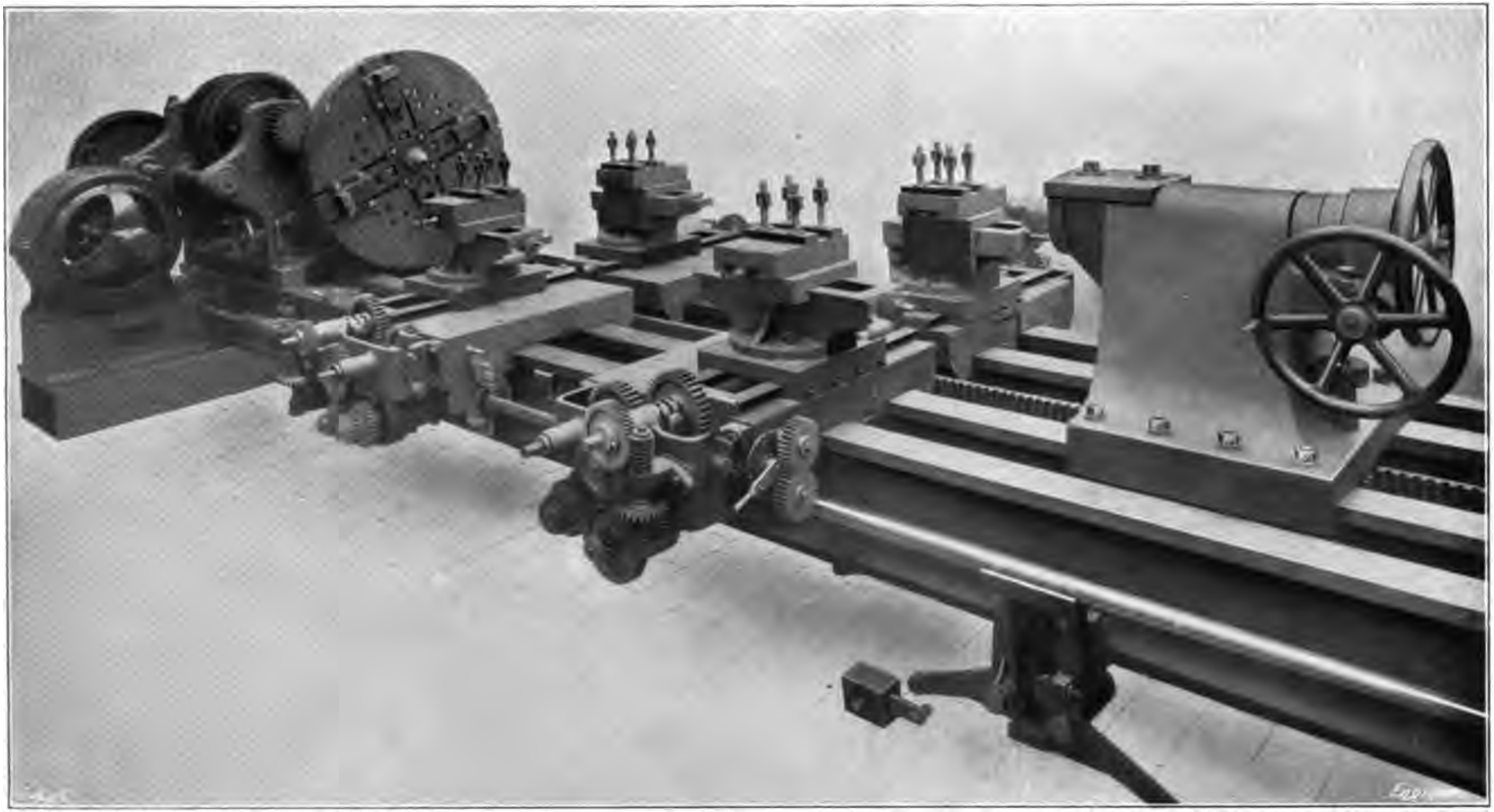


FIG. 180.—32-INCH CENTRE HIGH-SPEED LATHE (HULSE).

All feeding screws should be in tension for preference; of left-handed thread when the nuts move, and of right-handed when the screws move.

DEVICES FOR THE VERTICAL ADJUSTMENT AND CLAMPING OF THE TOOL

It is generally agreed that for good cutting the point of the tool should be raised above the centre of the work. When the tool is in this position, a given amount of top-rake, or a given cutting angle, leaves the tool of stronger shape than when placed below the centre. It is also not so liable to draw the rest forward, and so cause the tool to "run in"; for the horizontal outward component of the cutting force is then greater, and tends to spring rest and tool away from the job.

The trouble occasioned by the tool running into the work and chattering is more frequently the outcome of frail and springy rests, loose slides, etc., than to the position of the tool. For this reason all slides should be fitted with locking pads.

Packing strips are considered sufficiently good in this country for raising the tool to the proper position; whilst American makers provide wedge pieces, stepped washers, tilting rests, and the like.

The ordinary American rest (Figs. 179, A, and 183) permits of the tool point being raised; but as the single binding screw bears only on one point, little or no security is afforded to the tenure of the tool.

One of the best devices for adjusting the height of the tool is that of Lipe (Fig. 179, B), which consists of a circular-shanked tool holder let into a socket on the incutting slide, and locked to it by a split boss and tap screw. The tool is held squarely to the tool holder by two screws, and the whole can be bodily raised to the required position by another screw at the side.

Tool holders of the types shown in Fig. 179, A, lack the rigidity of the clamping plates, and the general convenience of handling of the British and Continental rests. A more satisfactory method than the latter for large lathes, at all events, has not yet been found.

It would appear to be advisable, however, in view of the number of tools and packing pieces to be stored away by the operator, for makers to adopt a standard dimension from the centre of the lathe to the top of the tool slide, for each size of machine. The number of steel sections to be kept in stock and the efficiency of the tool-room would thereby be increased.

The design of compound rests, shown in Figs. 154 and 155 (*ante*), and the front and back rests for the 32-in. centre lathe, shown in Figs. 180 and 181, are representative of first-class British practice for small and heavy lathes. The corresponding continental practice by De Fries (Fig. 182) and Ernst Schiess (Fig. 168, *ante*) show a close agreement in design and proportions.

These designs may be compared with that shown in Fig. 183 (see end of volume) by the Lodge and Shipley Company, of Cincinnati, U.S.A. It will be seen that even the best of American firms make their slides very light and narrow in comparison with similar-sized machines of European make. The base of the swivel slide, for example, of a 24-in. swing lathe (12-in. centres) of the former, is no greater than that of an 8-in. centre lathe of the latter. The holding-down bolts and width of other parts of the rests are in about the

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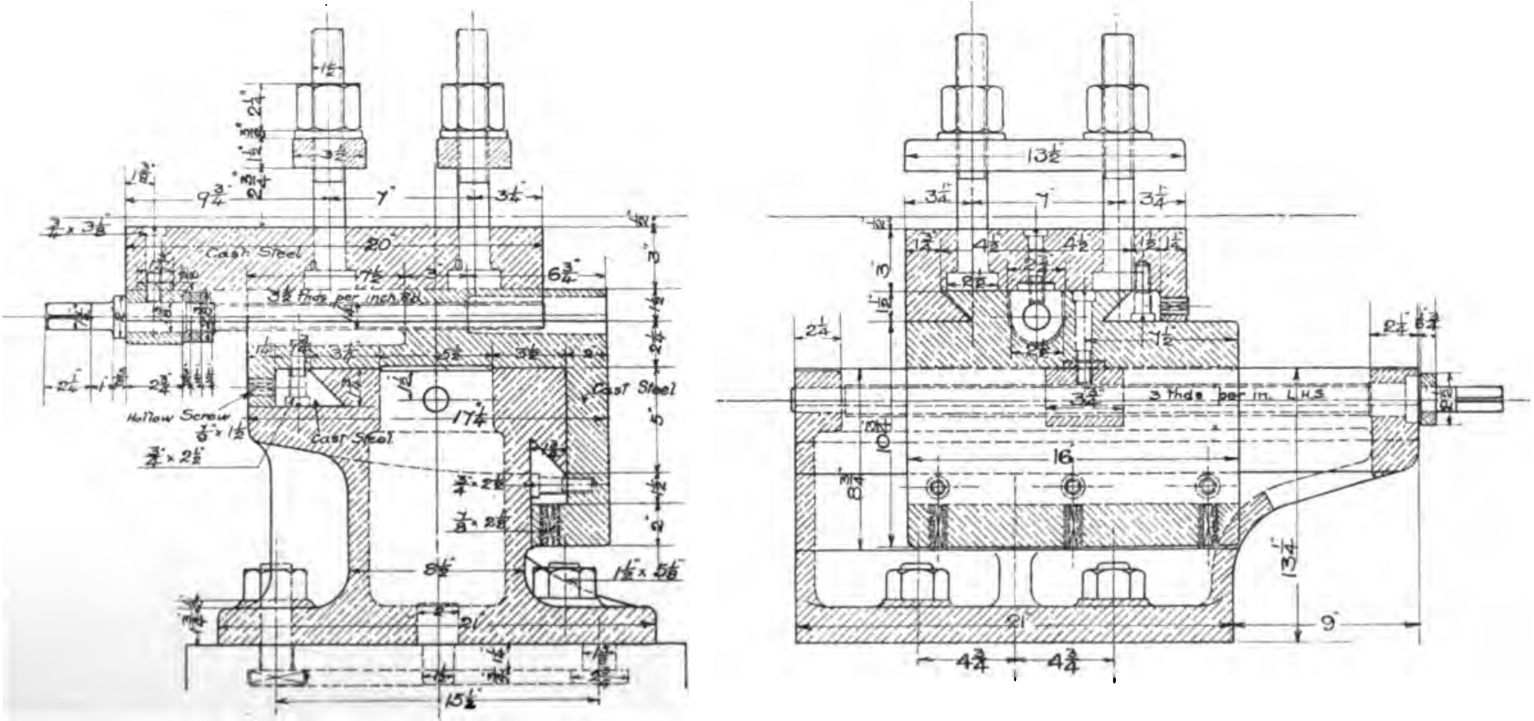


FIG. 181.—BACK RESTS FOR 32-IN. CENTRE LATHE (HULSE).

same proportion, so that the tool has very little chance of being rigidly held. Again, the rectangular tee slot for the reception of the tool holder in the American compound rest runs across the top slide at right angles to the ways of same. In order, therefore, that the tool and holder may not be pushed out, the operator brings the swivel slide round so that the slot is running in the same direction as the bed or work. This, however, robs him of the longitudinal screw adjustment by hand, and of the advantages of the compound rest, and he has two transverse movements (one being redundant), as the top slide then feeds in the same direction as the bottom or incutting slide; whilst the handles (which are fixed to the screws) usually foul each other.

Adjusting Strips for Wear.—If chattering is to be avoided, it is clear that whatever form of strip be adopted to compensate for wear, it must not impair the rigidity of the rest. Briefly stated, after the adjustment has been effected the strip or wedge must be solidly bolted to the slide, so that it becomes practically part of it. The usual arrangement of the slides is such that the thrust is taken by the solid strip, and acts normally to it, so that the force is distributed along the face. In the rests vee strips are generally employed, whilst between the saddle and bed vee and rectangular strips are both used; the former principally on small, and the latter on large lathes. A variety of such strips are shown in Figs. 184 and 185.

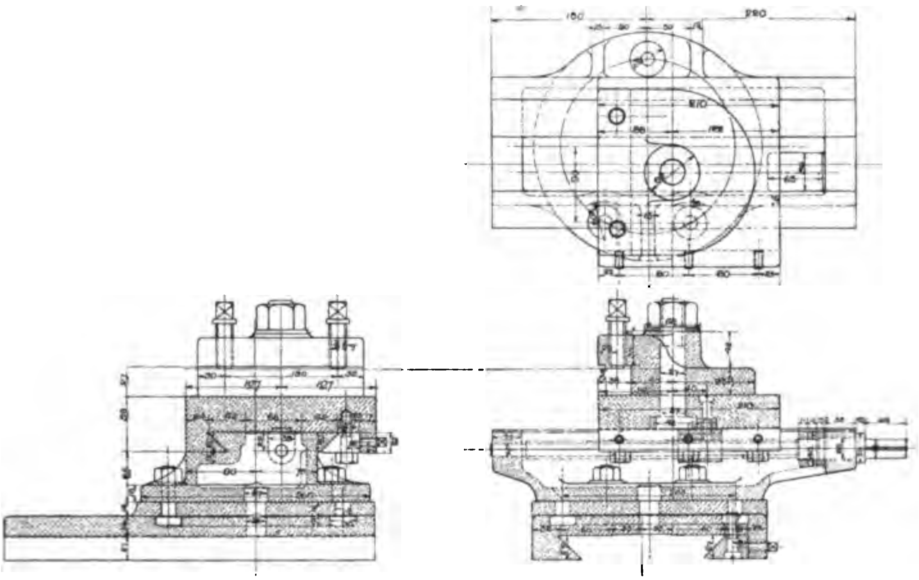


Fig. 182.—Compound Rests (De Fries).

As already mentioned, the thrust is usually taken on the solid vee, which brings the fitted strips to the front and right-hand side of the rests, where the adjusting screws are always accessible (see Fig. 184).

The form of vee strip most commonly used in this country is that shown at (A), where the strip is supported by vertical screws, and the adjustment effected by the horizontal screws only. Occasionally the latter are not readily got at, as in the saddle strip, for example; in that case the form (B) is frequently employed. Generally there is a considerable tilting action on the slides, which puts the vertical screws in tension, and they have not, therefore, the rigidity of those shown at (C), (D), or (E), where a metal-to-metal fit is secured. In order that the strips (C) and (D) may travel with the rests, they should be connected by a dowel pin; or the end of one of the pinching screws may be turned down to the bottom of the thread, and made to enter a recess in the strip. The design (E), although not quite up to the standard of the preceding examples, is an effective strip, and is to be preferred to some forms which have been adopted in cases when room was scanty. The vee wedge (F) forms a most substantial strip—affording a bearing solid with the rest, the entire length of the same. More nearly than any of the above it becomes essentially part of the rest; but it must be properly fitted to commence with. Should it only bear at the ends or centre, a wobbling or rocking movement will result, for the adjustment is effected by moving the

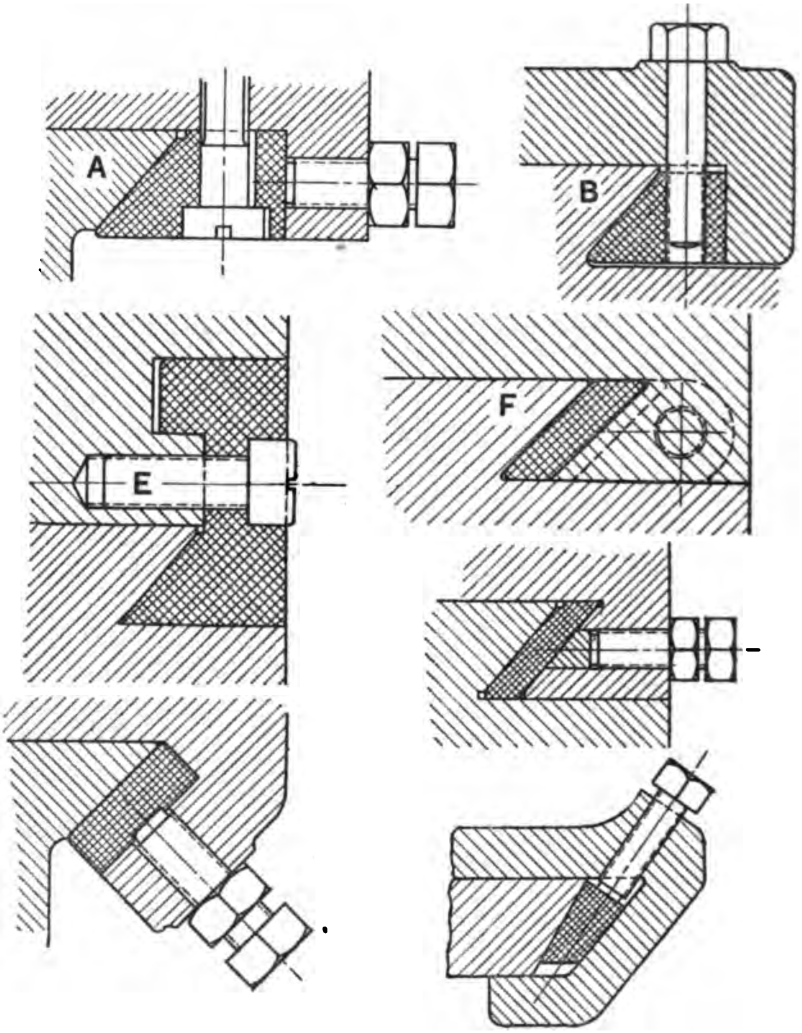


FIG. 184.

strip bodily lengthwise. This type, however, lacks the flexibility of the ordinary vee strip with the side adjustment, in that should the wear be greater at one end of the slide than at the other—as there is a tendency to do owing to the twisting action on the same—a variable compensation to suit cannot be made. The same objection might be raised to the rectangular wedge strip (G); but, judging by the frequency with which it is used, particularly between the bed and the saddle, such objection does not appear to be serious. The same class of strip is also used between the under side of the bedway and saddle to hold the latter down. More commonly, however, a plain rectangular strip is used, as shown at (G), as the wear in this direction is comparatively small. In the larger sizes of lathe, the length of the saddle becomes so great as to render the fitting and adjusting of the strips more difficult. As a consequence independent rectangular strips are much used to effect these adjustments. Many makers, however, use a plain tee or ell strip for this duty. Of these the ell strip, as shown at (H), is the more commonly employed.

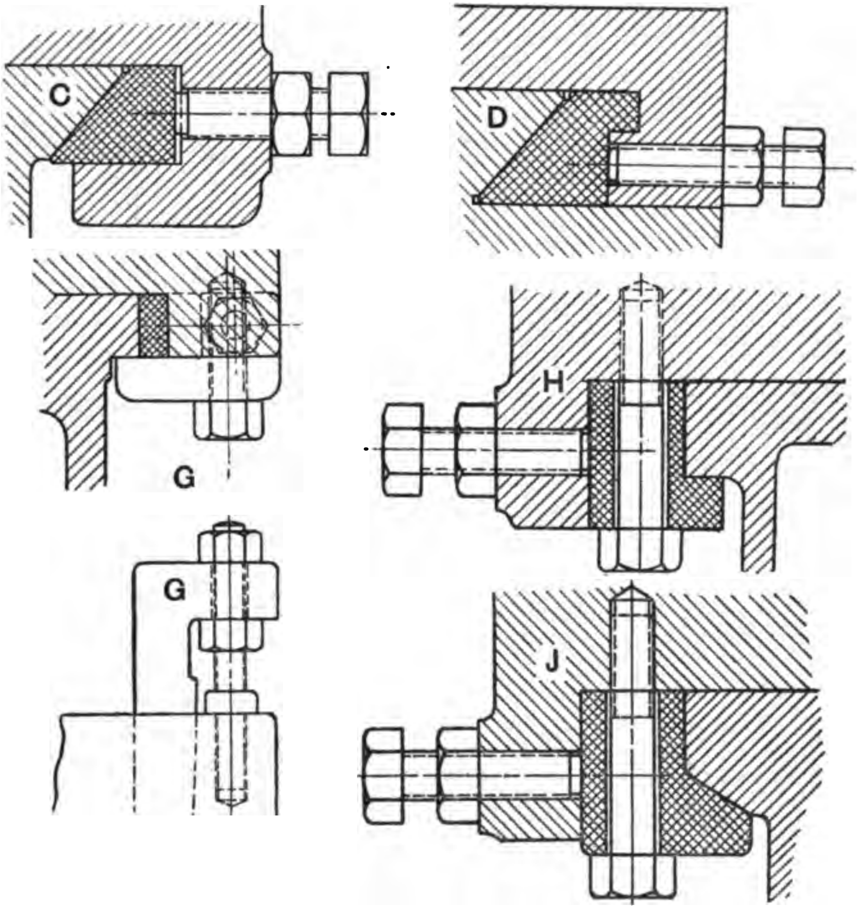


Fig. 185.

In Figs. 186 and 187 we present graphs of the proportions of the various parts of the slide rests. Fig. 186 gives those for lathes from 6-in. to 12-in.

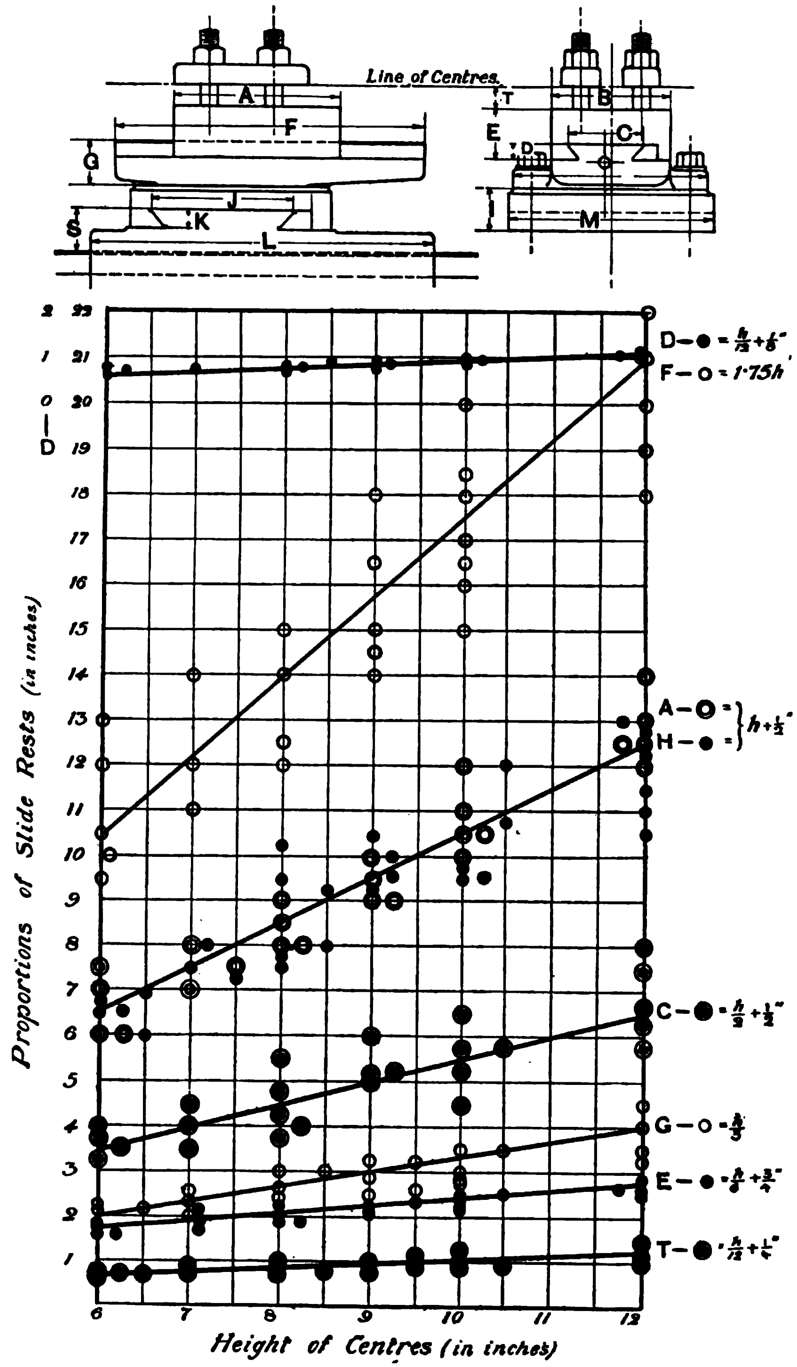

Line of Centres
A
F
G
J
K
S
L
T
B
C
E
D
I
M
Proportions of Slide Rests (in inches)
Height of Centres (in inches)
2 22
1 21
0 20
D 19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
6
7
8
9
10
11
12
D—● = $\frac{h}{12}+\frac{1}{8}''$
F—O = 1·75h
A—O =
H—● =
$h+\frac{1}{2}''$
C—● = $\frac{h}{2}+\frac{1}{2}''$
G—O = $\frac{h}{3}$
E—● = $\frac{h}{6}+\frac{3}{4}''$
T—● = $\frac{h}{12}+\frac{1}{4}''$

FIG. 186.

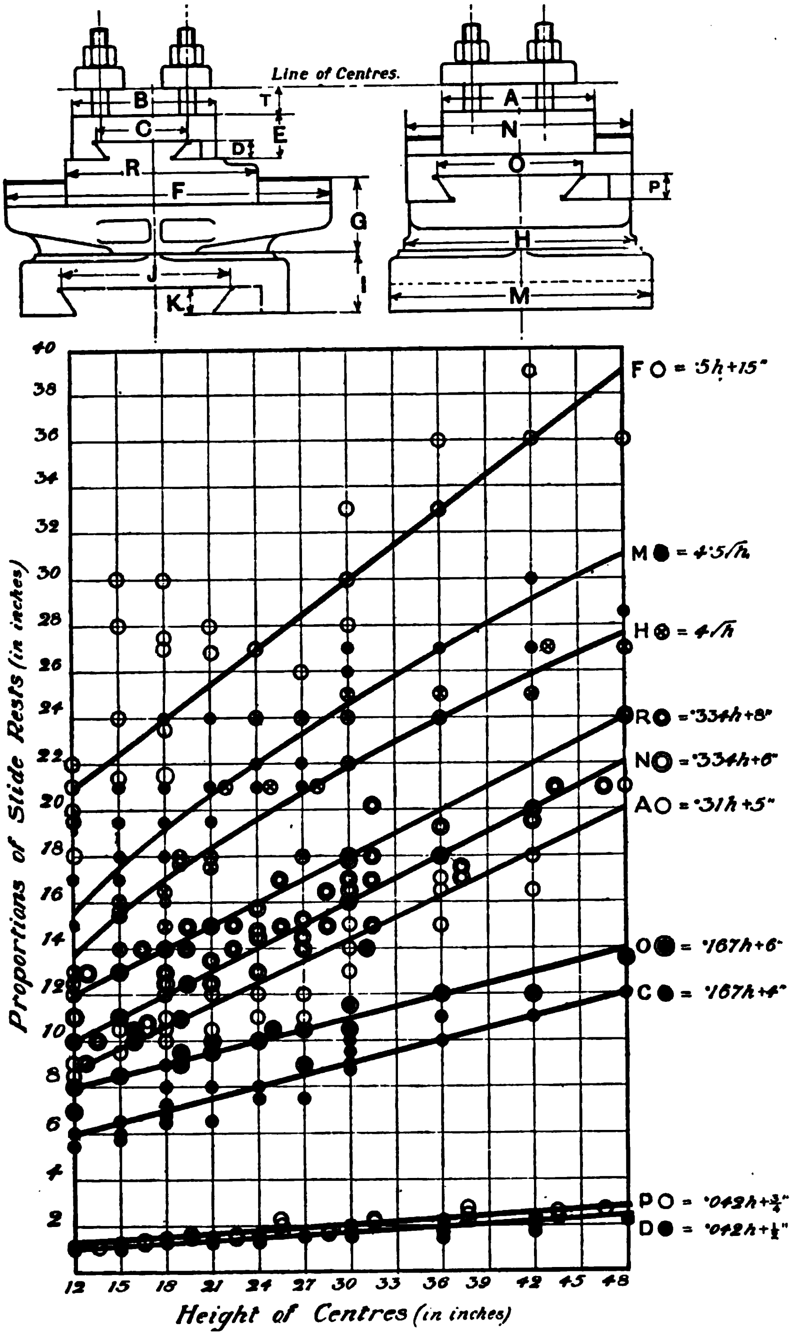

Line of Centres.
B
T
C
E
D
R
F
G
J
K
I
A
N
O
P
H
M
Proportions of Slide Rests (in inches)
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
12
15
18
21
24
27
30
33
36
39
42
45
48
Height of Centres (in inches)
F O = ·5h+15"
M ● = 4·5√h
H ⊗ = 4√h
R ◉ = ·334h+8"
N ◎ = ·334h+6"
A O = ·31h+5"
O ● = ·167h+6"
C ● = ·167h+4"
P O = ·042h+¾"
D ● = ·042h+½"

FIG. 187.

centres, whilst Fig. 187 shows the data for sizes from 12 in. to 48 in. The actual values plotted upon which these curves are based have been collected from the most recent practice. They apply equally to lathes for carbon steel or for high-speed steel.

The equations to the various graphs have been determined, and are given in Table LIX., herewith.

The sizes of the tool steel used in practice appear to follow a law $T = \frac{h}{12} + \frac{1}{4}$ in. As already mentioned, we should prefer to recommend the following values:—

For h = 6 to 9 in.	9 to 12 in.	12 to 15 in.	15 to 18 in.	21 to 27 in.	30 to 48 in.
T = 1 in.	$1\frac{1}{4}$ in.	$1\frac{1}{2}$ in.	$1\frac{3}{4}$ in.	$2\frac{1}{4}$ in.	3 in.

which would reduce the number of stock sections of tool steel to be maintained in shop, for the supply of a large number of lathes of various sizes.

TABLE LIX

Proportions of Compound Slide-rests for Lathes.

	Lathes up to 12-in. centres (Fig. 186).	Lathes from 12- to 48-in. centres (Fig. 187).
A	$h + \frac{1}{2}$ in.	$0{\cdot}31h + 5$ in.
B	$2{\cdot}5\sqrt{h}$	$2{\cdot}75\sqrt{h}$
C	$0{\cdot}5h + \frac{1}{2}$ in.	$0{\cdot}167h + 4$ in.
D	$0{\cdot}083h + \frac{1}{8}$ in.	$0{\cdot}042h + \frac{1}{2}$ in.
E	$0{\cdot}167h + \frac{3}{8}$ in.	$0{\cdot}083h + 1\frac{1}{2}$ in.
F	$1{\cdot}75h$	$0{\cdot}5h + 15$ in.
G	$0{\cdot}5h + 1$ in.	$0{\cdot}25h$ up to 30-in. centres
H	$h + \frac{1}{2}$ in.	$4\sqrt{h}$
I	$0{\cdot}22h + \frac{1}{2}$ in.	$0{\cdot}22h + \frac{1}{2}$ in.
J	$0{\cdot}8h$	$3{\cdot}2\sqrt{h}$
K	$0{\cdot}0625h + \frac{3}{8}$ in.	$0{\cdot}0625h + \frac{3}{8}$ in.
L	$8{\cdot}5\sqrt{h}$	$8{\cdot}5\sqrt{h}$
M	$h + 3$ in.	$4{\cdot}5\sqrt{h}$
N	—	$0{\cdot}334h + 6$ in.
O	—	$0{\cdot}167h + 6$ in.
P	—	$0{\cdot}042h + \frac{3}{4}$ in.
R	—	$0{\cdot}334h + 8$ in.
S	$0{\cdot}22h + \frac{1}{2}$ in.	$\sqrt{h}$
T	$0{\cdot}083h + \frac{1}{4}$ in.	$0{\cdot}05h + \frac{3}{4}$ in.

Dia. of screw in top rest	—	$0{\cdot}02h + \frac{3}{4}$ in.	Solid screws in tension.
" " swivel slide	$0{\cdot}08h + \frac{3}{4}$ in.	$0{\cdot}03h + \frac{3}{4}$ in.	
" " saddle	$0{\cdot}05h + \frac{3}{4}$ in.	$0{\cdot}05h + \frac{3}{4}$ in.	
" " clamping bolts	$0{\cdot}025h + \frac{3}{4}$ in.	$0{\cdot}025h + \frac{3}{4}$ in.	

CHAPTER XXXVI

STEADY RESTS

Travelling and Fixed Stays.—When the work is small in diameter and comparatively long, it is necessary to have a support to the same in addition to that provided by the fast and loose headstock. The work may be of such a springy character as to necessitate a support immediately above and behind the tool. For this purpose a stay, mounted on the saddle, and travelling with it, is usually supplied. The unstayed portions of the work between the tool and the centres, if unsupported, may also sag, and set up a whirling action which tends to throw the work out of the centres, and to induce vibration in the machine. In that case, one or more stays fixed to the bed at intervals along the work are required. Again, when work rotated in the faceplate has to be drilled or bored, the loose headstock is not available as a support. It is often then employed in feeding the drill or bit forward. A fixed stay is then requisitioned to uphold the job, in order that the overhang may be reduced and the required rigidity secured. When the job is black or rough it is impossible to make it run truly in the support. A round muff is therefore slipped over and held fast to it by pinching screws, so that the muff carrying the work and running centrally then rotates in the stay.

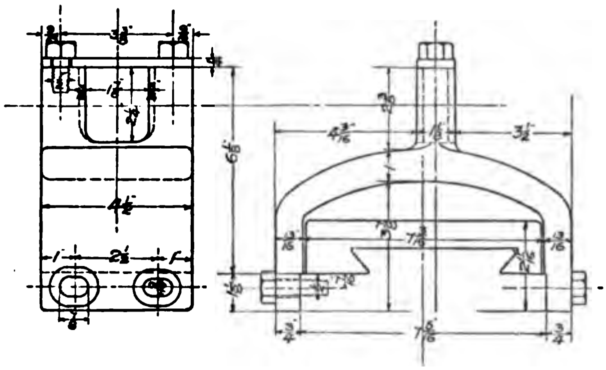


Fig. 188.—Travelling Stay (Lang).

The primary function of the stay is to provide a substantial support to the work; but it must also be capable of ready adjustment to suit varying diameters, and of moving clear of the work altogether to permit of its being withdrawn without disturbing the setting. This is particularly necessary for rapid production and repetition work.

An early design of fixed and travelling stay, and one still largely used, consisted of a cast-iron stand having a base to suit the bed ways when fixed, and the saddle ways when travelling. The saddle wings were occasionally provided with tee slots (a practice which is continued on the Continent), upon which the travelling stay was fixed. An arm bolted to the stand, and fitted with two half bushes of hard wood (sometimes these were lined with tin or sheet metal), held in place by a cap, received the work. A similar device is shown by Fig. 188. This form of stay may no doubt be advantageously used on parallel shafts, etc.; but it is quite unsuitable for stepped

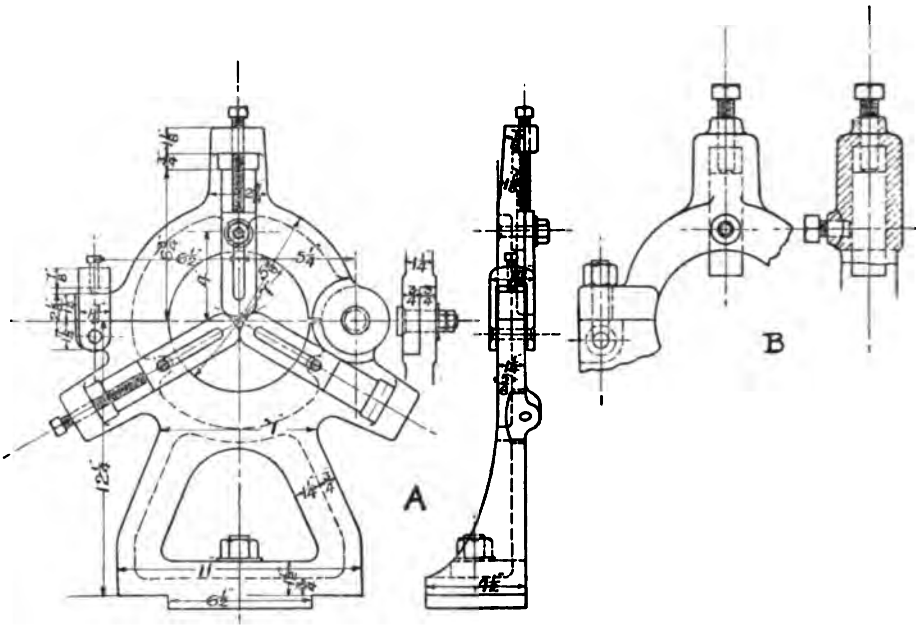


Fig. 189.—Fixed Stay (Richards).

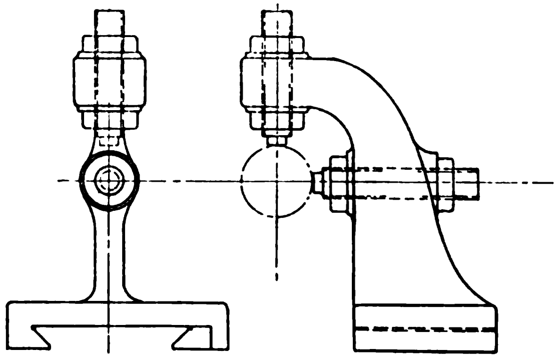


Fig. 190.—Screwing Stay.

work, as the bushes have to be changed for practically every diameter. The tendency of the bushes to misplacement, and the amount of time spent in looking for them, as well as the lack of convenient adjustment, are serious drawbacks to this design.

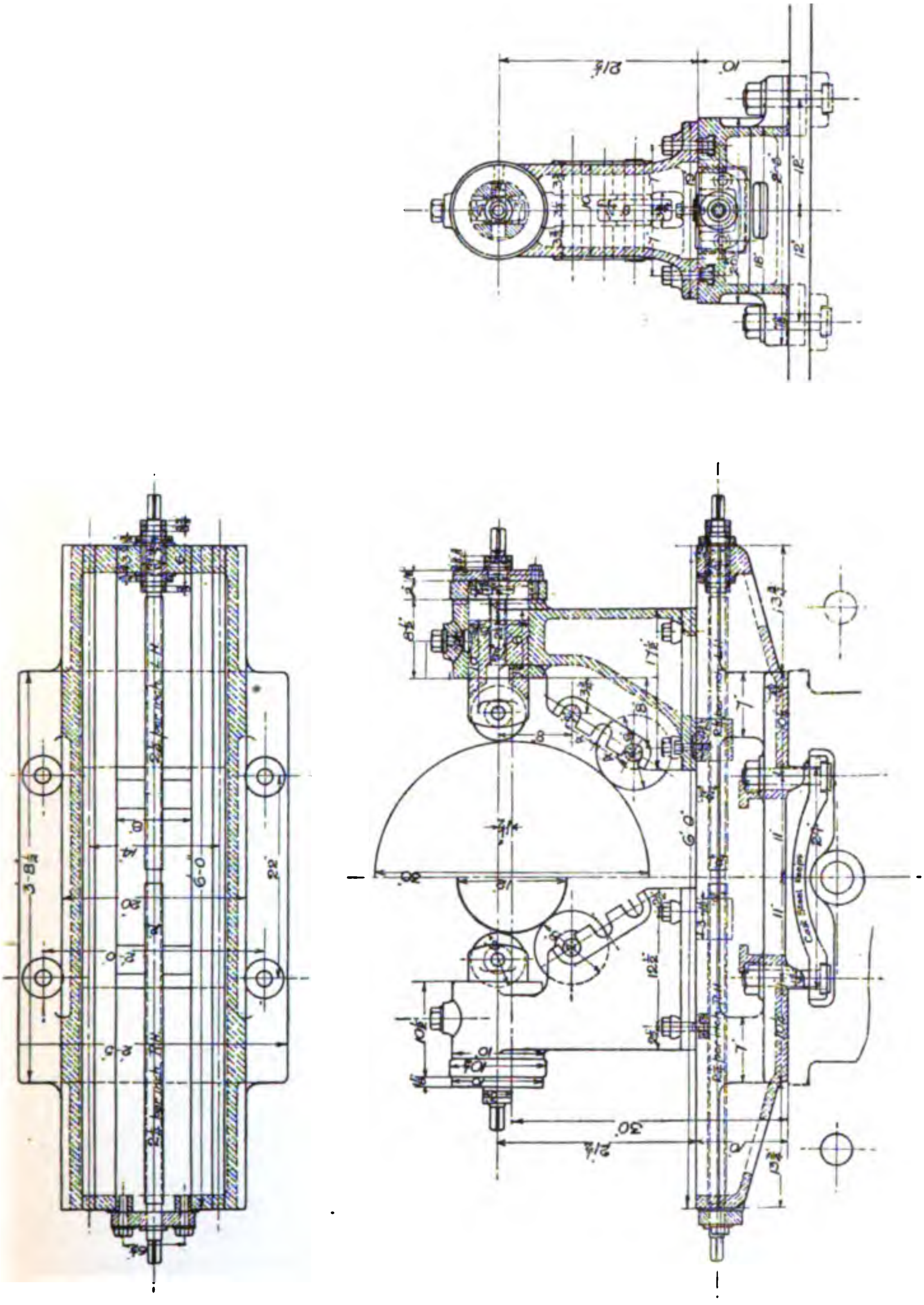


FIG. 192.

A more commendable form is shown in Fig. 189, A, where the work is supported on three adjustable hardened steel jaws fitted in the radial slots of the stand and hood. Each jaw is thrust forward by a set screw acting on its end, and is locked when in position by another screw which fits the slot in the centre of the same. The hood carrying the top jaw is hinged at the right hand, whilst at the opposite side a hook bolt or stirrup piece is provided to hold this section in position. The jaws are occasionally made of round steel, and slide in circular guides as shown in Fig. 189, B. This forms a most substantial rest and is very cheap to manufacture.

With this design of stay the hood can be thrown back, to allow of the work being removed; or it can be brought into place again without disturbing the setting. Although the example shown in the figure represents a fixed, or stationary support, the same design may also be used for travelling; the wings of the saddle being then prepared to receive it. In that case, however, the stay is at some distance from the tool, and very often in the way of the swivel slide; or it may necessitate a larger overhang of the tool than is desirable.

An excellent example of a three-jaw travelling rest is shown in Fig. 161 (see end of book), where it is mounted on the transverse ways of the saddle. The position of the jaws has been altered and the stay being open at the front does not in any way obstruct the slide rests. The hinged hood is retained and the position of the locking bolts to same ensures a most rigid support.

Another type of travelling stay much used for screw-cutting work is shown by Fig. 190; and consists of a stand with two adjustable studs, hardened at the ends, upon which the job bears.

An original design of a follower rest is shown in Fig. 191 (see end of book). It is especially convenient when turning work having a number of shoulders on it.

Two jaws carrying hardened-steel rollers move in and out in a circular path, being actuated by a worm and knob. When set in any position, they are adapted for a variety of diameters by simply moving the entire stay backward or forward. This is accomplished by connecting the stay to a screw which telescopes through the regular cross-feed screw, and is operated by the same hand wheel which sets the tool rest. The position of the rollers is such that in approaching a shoulder they support the shaft upon the smaller diameter, until the cutting tool has turned a portion of the next larger diameter, when the position of the rest is changed to bear on that portion.

Fig. 192 presents another form of roller stay for heavy work, by Messrs. Hulse. The two top rollers are placed above the centre, and are carried by a sleeve, adjusted by screws mounted in the uprights. The uprights are stepped to receive the bottom rollers, and are simultaneously moved in or out by the right- and left-handed screw in the bottom slide. For the class of work mentioned this forms a first-class stay.

CHAPTER XXXVII

LOOSE HEADSTOCKS

THE function of the loose headstock is twofold. In addition to providing a support to the end of the job it must be capable of a delicate adjustment to compensate for the variation in the length of the work, which is occasioned by rise of temperature. The longitudinal movement is also required for setting up the work; and, as previously mentioned, for feeding a drill or bit forward when drilling or boring. For this reason it is usually fitted with a non-rotating spindle, which is advanced or withdrawn by means of a screw and nut, and locked whilst in position by suitable means. The work, therefore, rotates on a dead centre. The fast headstock centre, on the other hand, rotates with the work, and there is no relative rubbing between them.

It is believed that work produced on live centres is less accurate than on dead centres, as in the case of grinding machine and axle-turning lathes. In the former the rubbing speed is high; but the force exerted on the work and centres is comparatively small. In the latter case the work is of short length and stiff, being rigidly held at the middle. This considerably reduces the force that would otherwise come on the centres. Expansion, due to change of temperature, in this case is also almost negligible, as the axle is only machined for a short distance at each end.

In grinding machines a strong spiral spring is fitted at the back of the headstock spindle, or some equivalent means is provided to allow for any change of length.

The use of a fixed centre allows of a minimum side play, and it is therefore possible to obtain more accurate work with it than with a live centre. On the other hand, the small bearing surface of the work upon the fixed centre with its accompanying high bearing pressure, notwithstanding its relatively small rubbing speed, frequently causes seizing. This is only prevented by the constant attention of the operator.

Several schemes have accordingly been devised for rotating the loose headstock centre with the work. In some cases the centre and spindle rotate together, as in the wheel lathe. In other cases, again, the centre is mounted in a ball race fixed in the non-rotating spindle. The former is the more reliable, but the expense of its manufacture is against its general adoption.

The reason for its introduction in the case cited is that the power would otherwise require to be transmitted through the axle to the wheel at the loose headstock. As, however, the cuts are usually heavy, and beyond the transmitting capacity of the axle, a faceplate is mounted on each headstock, and motion simultaneously communicated to these, so that the axle is unstrained.

A certain small amount of side-play is unavoidable in ball bearings, and this has been against their adoption for journals. The same defect appears also in the designs of centres so mounted. Even in comparatively small lathes, the ball-bearing centre has not proved sufficiently successful to be generally adopted. One firm (Tangye; see the *Engineer Supplement* for November 4, 1904) tried a cup centre of 60°, into which was placed a ball, and which also partially entered the recess in the work. With light work the ball went round with the work, and proved fairly satisfactory; but when tried with heavy work the ball stuck fast, and soon began to heat up.

Few people appear to recognize that much of the trouble attributed to the dead centre is due to the increased thrust which is brought upon the same by the expansion of the work. When the job is put between the centres, they are jammed hard up, and a subsequent expansion of the work therefore increases the load on the thrust bearing of the fast headstock and upon the dead centre. Now, there appears to be no reason why a chuck should not be freely mounted on the barrel nose, so that it would rotate and be driven by the work. If the thrust bearing in the fast headstock be properly designed and constructed, there is no objection to the above chuck having a little end play to allow for expansion.

This may be objected to on account of the limited bearing surface afforded, or the large overhang necessary to obtain the desired surface. This in turn might be easily overcome by providing a sleeve free to rotate in the barrel of the head and the chuck mounted in the end thereof. To support the work whilst the end of the job, and that part occupied by the jaws of the chuck, are being machined, the ordinary non-rotating and adjustable spindle carrying the centre could be fitted within the rotating sleeve, and be brought into use for this purpose, and also for centering. The centre spindle would then be capable of advancing in the ordinary way to allow for feeding drills, etc. The sleeve and centre spindle in the above arrangement would be practically identical to that employed in the ordinary drilling machine.

Regarding locking devices to the spindle, several designs are shown in the accompanying figures. It is necessary that, if the work is to be held rigidly, there must be no spring of the spindle, and all split-cone or lug-locking arrangements are unsuitable for the purpose. The remarks made upon the use of the split cone for the fast headstock spindle apply equally in this case.

The spindle must be a good fit in the barrel (*i.e.* ground into place) to begin with, and locked by a screw and pad.

The holding-down bolts should be short, so as to diminish as much as possible the lift due to stretching, and placed as far apart as may be, both lengthwise and crosswise, to prevent tilting.

To secure a more powerful nipping action, the screwed ends of the holding-down bolts should be made considerably larger in diameter than the body of the bolt (the latter being, of course, maintained of the size now usually fitted, say, $\delta = \frac{1}{3}\sqrt{h}$), screwed with a fine thread and fitted with a deep nut. In this way the steel bolts could be stressed nearly to the elastic limit when screwing up, and the additional load afterwards imposed on them by the cutting force would have an inappreciable effect in lengthening them. The loose headstock would then be so firmly held to the bed that other mechanical means, such as the pawl and rack, for preventing it from slipping back would probably be unnecessary.

An exhaustive discussion of the straining actions induced in the mandrel of the loose headstock would lead us to very complex expressions, which would be practically valueless; but the following simple treatment gives formulæ agreeing remarkably well with practice.

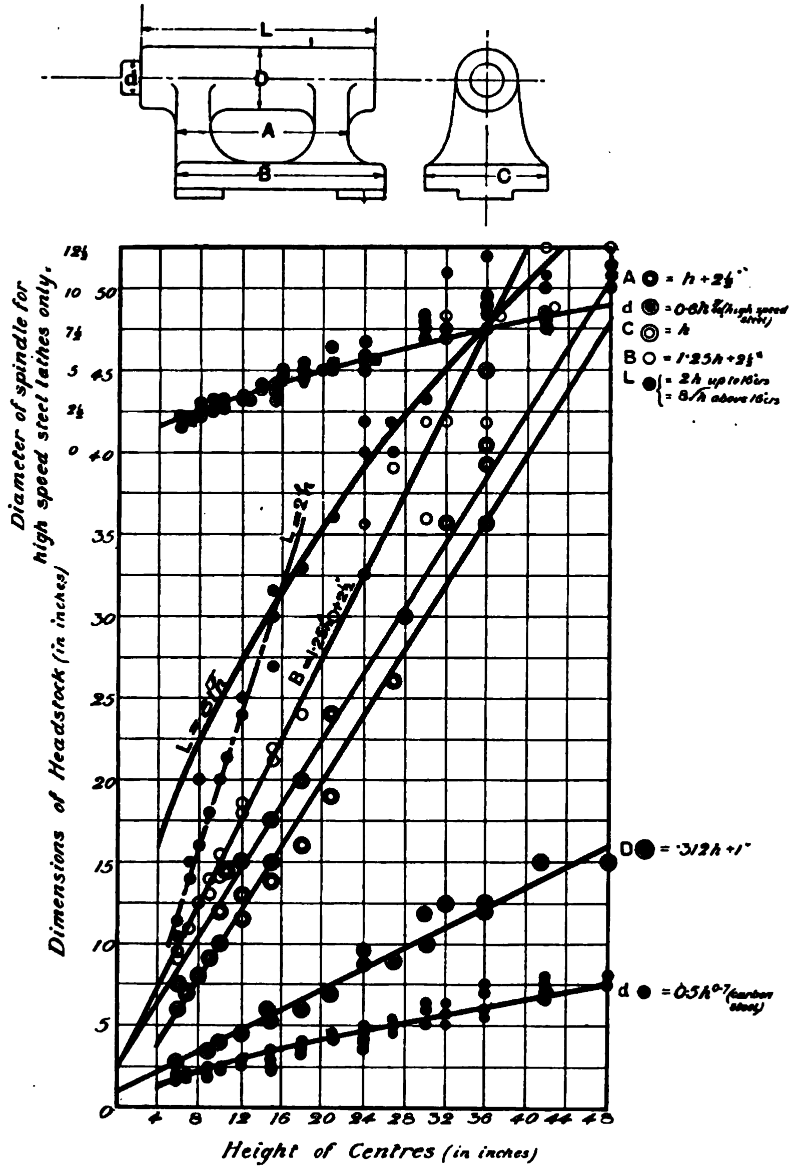


FIG. 193.

When the thrust due to the cut, $F = \frac{100h^2}{6400} = \frac{h^2}{64}$ tons, is acting near the loose headstock, let the overhang be $l_1 = \sqrt{h}$. The length of the barrel being about $8\sqrt{h}$ (*vide* Fig. 193), the distance to the point of application of the reaction Q from the barrel end B will be (about) $l_2 = 7\sqrt{h}$.

The deflections of the mandrel at the places where F and Q act will be given by—

$$\delta_1 = \frac{Fl_1^3}{8EI} \text{ and } \delta_2 = \frac{Ql_2^3}{8EI} \text{ respectively} \quad . \quad . \quad . \quad . \quad (149)$$

Now $$Fl_1 = Ql_2 \therefore \delta_2 = \frac{Fl_1l_2^2}{8EI}$$

The absolute amount of deflection allowable at Q will depend on the original fit of the mandrel in the barrel, and the slackness of the fit will be about proportional to the diameter of the mandrel. We may therefore take $\delta_2 = c\text{D}$ where D is the mandrel diameter. With E = 12,000 tons per square inch, and $\text{I} = \frac{\pi \text{D}^4}{64}$ we obtain

$$c\text{D} = \frac{h^2 \times \sqrt{h} \times (7\sqrt{h})^2 \times 64}{64 \times 8 \times 12{,}000 \times \pi \text{D}^4}$$

$$\text{or D}^5 = \frac{49}{64 \times 8 \times 12{,}000 c\pi} h^{\frac{7}{2}}$$

$$\text{or D} = \text{K}h^{0\cdot7} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (150)$$

With K = 0·5 for carbon steel, and K = 0·6 for high-speed steel lathes, we obtain the curves shown on Fig. 193.

The deflection which this method of calculation, applied to the practically obtained data, and found to agree therewith, allows is about $\frac{1}{500}$th of the diameter of the spindle.

Fig. 193 is the result of the compilation from practice of the various principal dimensions of loose headstocks. Fair curves have been drawn through these, and the equations to these are given in the Table LX. below.

TABLE LX

Proportions of Loose Headstocks

Dimension on figure.	Value in terms of h.
A	$h + 2\frac{1}{2}$
B	$1\cdot25h + 2\frac{1}{2}$
C	h
d	$0\cdot5h^{0\cdot7}$ (for carbon steel lathes)
d	$0\cdot6h^{0\cdot7}$ (for high-speed steel lathes)
D	$0\cdot312h + 1$ in.
L	$2h$ (for lathes up to 16-in. centres) $8\sqrt{h}$ (for lathes above 16-in. centres)

Dia. of holding-down bolts	$\frac{1}{3}\sqrt{h}$			
Number of bolts	6 in. to 12 in. 2	12 in. to 24 in. 4	24 in. to 36 in. 6	above 36 in. 8

LOOSE HEADSTOCKS FOR SMALL LATHES

A good example of a modern design of headstock for a small lathe by Greenwood and Batley, Ltd., is shown in Fig. 194. Here the headstock is arranged for taper turning. The bottom plate fits between the shears, and is furnished with a fixed nut, which receives the cross-adjusting screw in the headstock proper. The latter is checked into the bottom plate to prevent longitudinal movement, and the whole is bodily held to the bed by the two holding-down bolts which pass through the elongated slots in the bottom plate to the clamping plate. A plug is provided for locating the true central position of the headstock. The adjustable spindle is hollow, and fitted with a gun-metal nut to receive the actuating screw, and is locked when in position by a bolt and split-lug device.

Another example for the same size of lathe, also fitted with taper attachment, is shown by Fig. 195. It is by John Lang and Son. The centre bolt binds the plate and headstock together, whilst the front and back bolts lock the headstock through the long clamping plate to the bed. This arrangement allows the headstock to be moved along the ways without disturbing it when set.

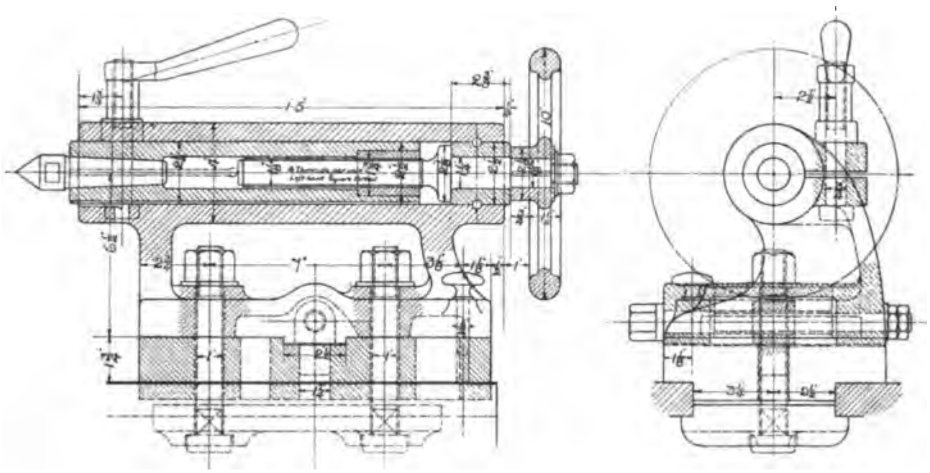


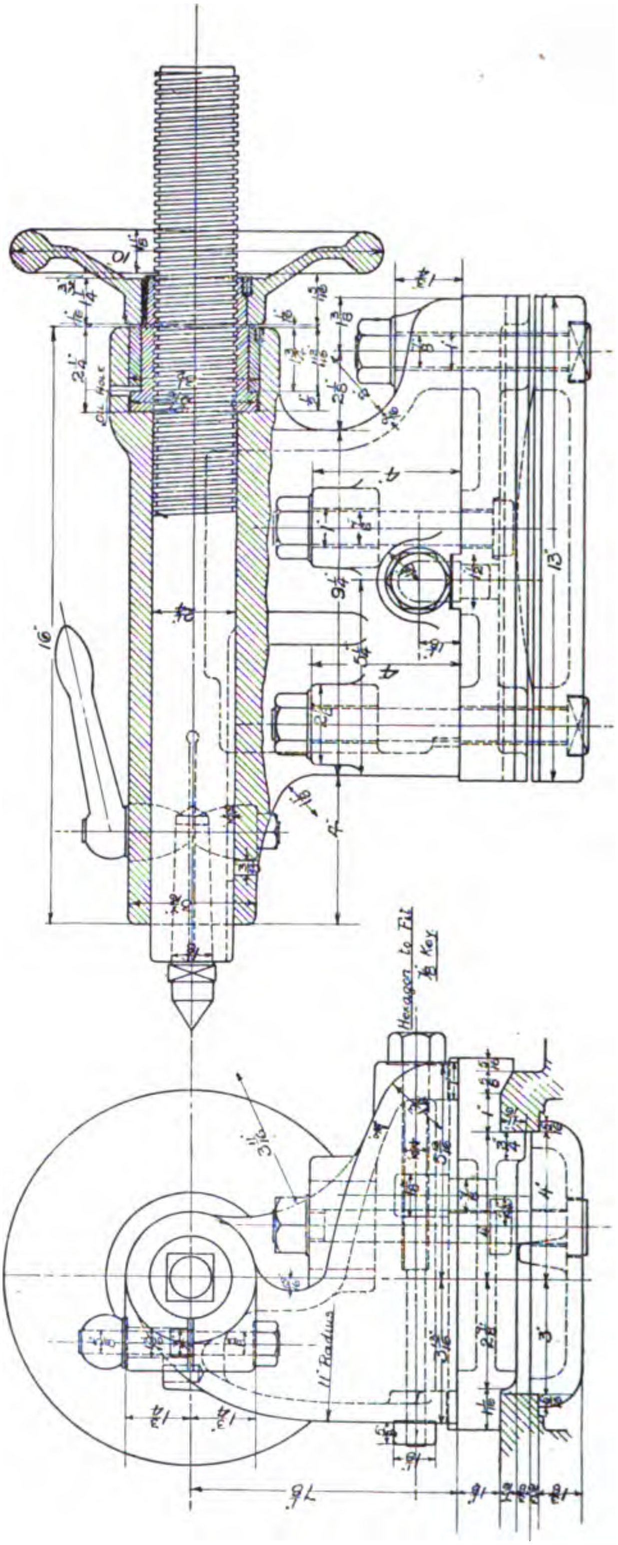
FIG. 194.—8½-IN. CENTRE LATHE LOOSE HEADSTOCK (GREENWOOD AND BATLEY).

The sole plate is made to fit on the top back surface, and on the vertical and slant faces of the inside surface of the bed, but is just free of the latter top surface. Since the saddle does not bear on any of these parts, the alignment of the centre is not disturbed by wear due to the saddle. The mandrel and screw are in one piece, and the top of the screwed part is made to fit the barrel, the object being to obtain a length of bearing equal to the full length of the barrel even when the spindle projects a considerable distance.

Unless the spindle is made of a correspondingly larger diameter, this advantage will not be realized, however; for the amount of slack originally necessary will permit of deflection taking place within the barrel, and the spindle will bear on the barrel not very far from the front end.

SELLERS' LOOSE HEADSTOCK

In the loose headstock shown in Fig. 196, by William Sellers and Co., the spindle is advanced in the usual way by the hand wheel and screw; but



8¼ inch Centre Lathe Loose Headstock.
(Lang)

FIG. 195.

the locking device to this spindle is obtained through the concentric split cones A and B. The rear of cone A is threaded to receive the combined nut and sleeve, and a handle fitted to the sleeve rotates the nut, which thrusts against the reverse split cone B, free on A, thereby thrusting it against the barrel, and concentrically locking the spindle at two places, viz. the end and middle of the barrel. When unlocking, the nut thrusts against the screw-thrust block, and pushes A forward. The headstock is gripped to the bed by four bolts and a clamp plate the full length of the base. The bolts, on being tightened, pull the clamp plate up the inclined surface beneath the back way of the bed, whilst a vertical face on the clamp plate bears against the headstock base, thus forcing it against the vertical face on the bed, and preserving the vertical alignment of the head.

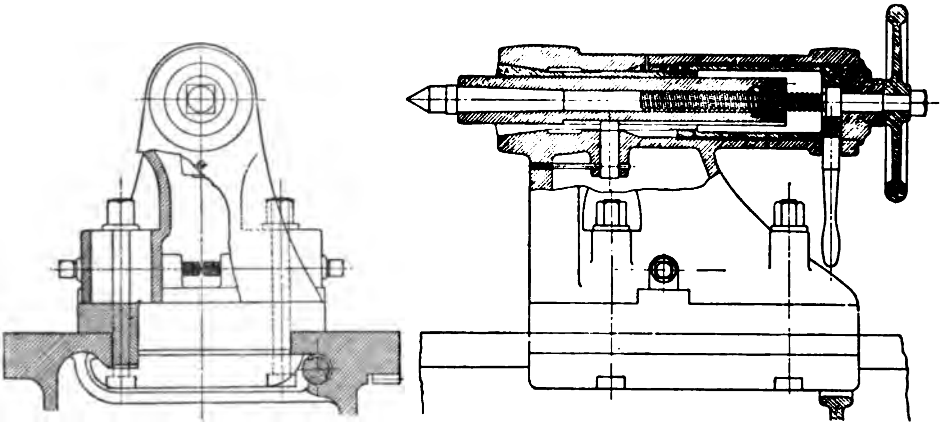


Fig. 196.—Loose Headstock (Sellers).

An old, and what appears to be a very suitable locking device is shown in Fig. 197.

Here two hollow cylinders are pulled hard against the spindle. The backward thrust is relieved by nipping the mandrel between these cylinders and the top of the barrel.

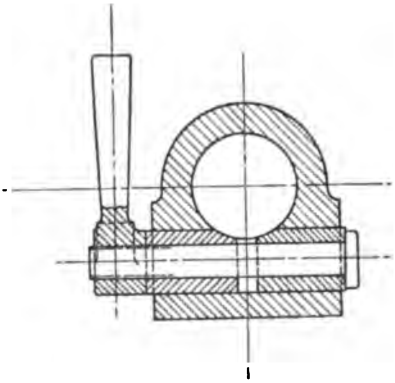


Fig. 197.—Locking device.

It is difficult to know why this device should have been discarded for the split-lug arrangement now so commonly in vogue. If the mandrel is a slack fit in the barrel, it will be pushed out of centre by this arrangement; but there is no reason why this slackness of fit should exist. In the ordinary split-barrel gripping device the positions of the contact arcs of the barrel upon the mandrel altogether depend upon the shape of the former, and it by no means follows the process of nipping up a split barrel leaves the alignment of the mandrel unaltered.

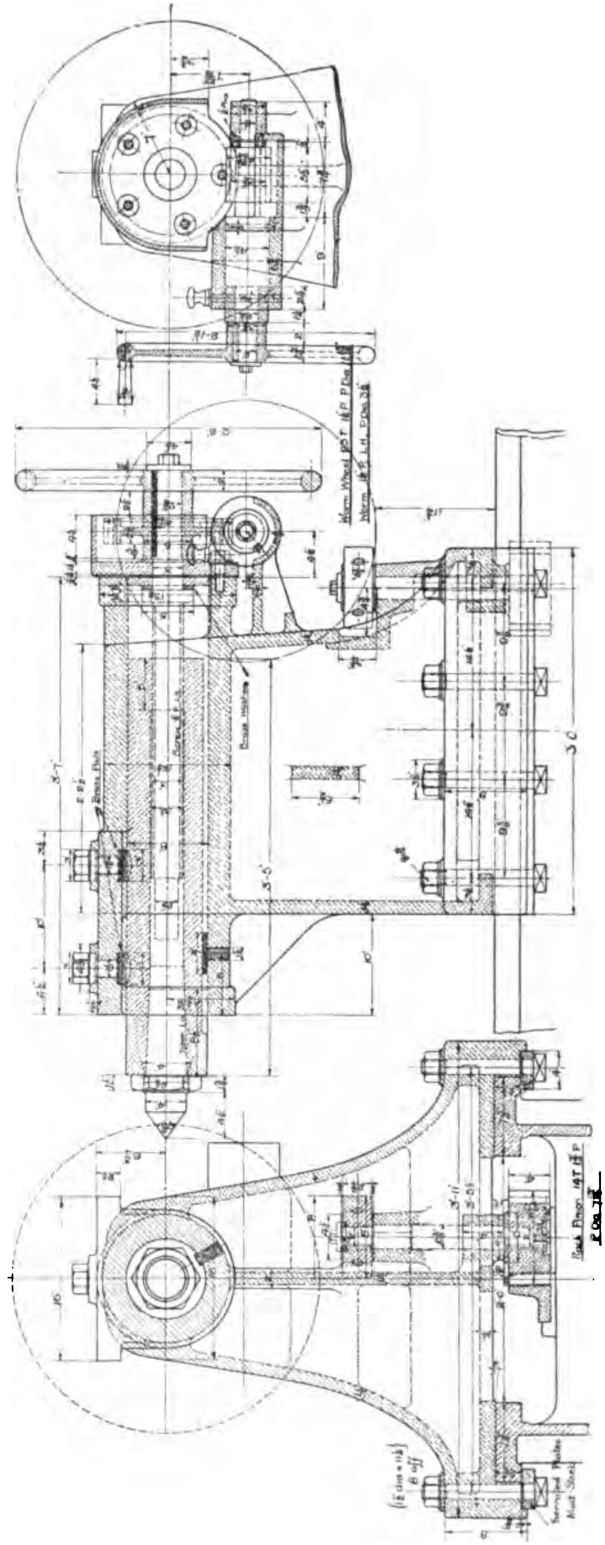


FIG. 198.—LOOSE HEADSTOCK FOR 32-IN. CENTRE LATHE (HULSE).

Loose Headstocks for Large Lathes

In the case of large lathes, no such splitting of the barrel is admissible. When the forces due to cutting become great, the resulting stretching of the nipping screw increases so much as to allow of the alignment being vitiated, and chattering ensues.

Fig. 198 shows a modern loose headstock for a 32-in. lathe by Messrs. Hulse. This headstock is fitted with a spindle 8-in. diameter, which can be quickly advanced or withdrawn by a hand wheel on the screw. A worm wheel is also keyed to the screw for finer adjustment and more powerful feed. This wheel is covered by a guard, and is combined with the thrust block. The worm can be quickly withdrawn, and runs in an oil bath. A rack situated within the bed, and pinions carried by the head, are employed for shifting the head along the bed. The headstock is held to the bed by serrated mild steel plates and eight 1½-in. bolts.

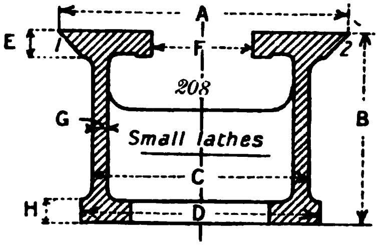
A loose headstock for a 1500 mm. (60 in.) centre lathe has already been illustrated (Fig. 168; see end of book). It can be set over for taper turning, and is fitted with a spindle 225 mm. (9 in.) diameter by 1780 mm. (75 in.) in length. The adjustment to the spindle is made by a hand wheel placed at the front of the headstock, and suitable mitre and spur wheels. By having the hand wheel placed thus instead of at the rear of the headstock, the operator has the centre more immediately in view when setting up the work.

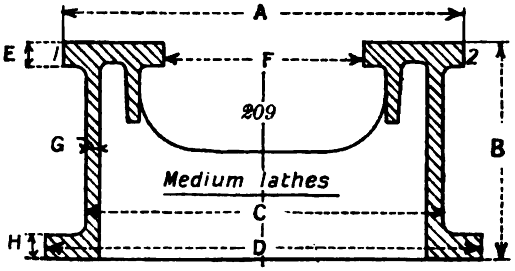
The headstock has a support on all four ways of the bed, and is bolted to it by eight 50 mm. (2 in.) bolts. A positive locking device, consisting of a pawl and rack fitted to the bed, prevents it slipping back whilst cutting. This headstock can also be traversed along the bed by means of a hand wheel at the rear, a quadruple threaded worm which gears with a 30-toothed wheel, and a pinion which meshes with a vertical rack at the centre of the bed. To facilitate hand traversing, four rollers—two at the front and two at the rear—are mounted excentrically in the headstock base, and can be quickly put in or out of action.

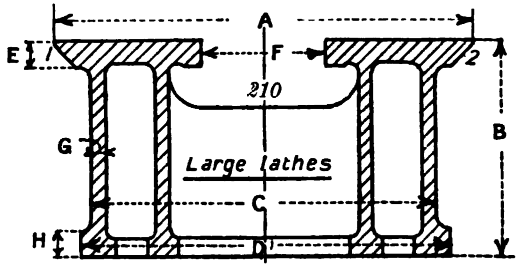
CHAPTER XXXVIII

LATHE BEDS

The forms of section of bed most usually employed in this country—when with two ways only—are shown by Nos. 208, 209, and 210.







No. 208 is used for lathes from 6-in. to 18-in. centre.
No. 209 „ „ „ 18-in. to 30-in. „
No. 210 „ „ „ 30-in. upwards.

The chief or fundamental dimensions of the sections, as indicated by the letters on these three figures, have been plotted in Fig. 230 on a base of height of centres. The following expressions have been deduced from this figure, and give the relations of these principal sizes to the size of the lathe.

$A = 1{\cdot}875h$ to $2h$.
$B = 4\sqrt{h}$.
$C = 1{\cdot}5h$.
$D = 1{\cdot}5h + \frac{2}{3}\sqrt{h}$.
$E = \frac{h}{61} + \frac{3}{4}$ in.
$F = \frac{3}{4}h - 2$ in.
$G = \frac{1}{3}\sqrt{h}$ if one web per side.
$G = \frac{1}{4}\sqrt{h}$ if two webs per side (*i.e.* inverted U).
$H = \sqrt{h}$.

These rules are applicable for lathes below 36-in. centres, and with two ways only.

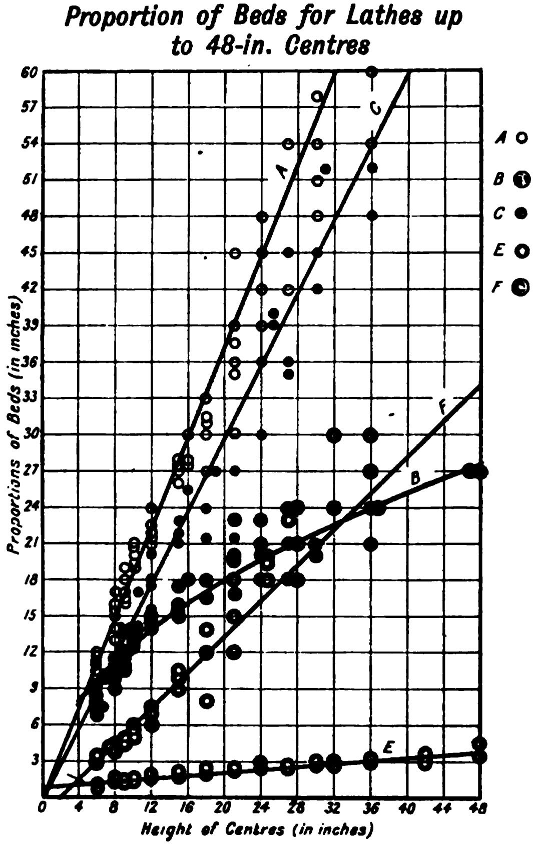

Proportion of Beds for Lathes up to 48-in. Centres

FIG. 280.

The figures throughout give transverse sections of the beds as seen from the fast headstock end, so that the front of the machine is at the right hand of the figure.

Form of Guiding Surfaces.—In most lathes of British design the top surfaces of the ways are flat—*vide* Nos. 208 to 223. For the side guides, inclined surfaces upon both the back and front edges of the bed have been employed for all sizes of machine—*vide* No. 208 (1) and (2). The inclined guiding surface on the edge of the bed was found to be a very convenient form, as the backward thrust of the tool had the effect of pulling the saddle to the bed. Similarly, a simultaneous downward and forward adjustment was effected by pinching up the vee strip, usually placed at the front of the saddle. An objection has been raised to the back inclined surface on account of the horizontal outward force on the tool producing a jamming tendency between the saddle and the bed. As a consequence, many lathes are fitted with a flat surface at the back to take the thrust, whilst the vee strip at the front is retained for adjustment.

The placing of these guiding surfaces on the extreme outside ways, and not upon the front way only, no doubt tended to economy in construction, and certainly secured the saddle against tilting. When the guiding surfaces are confined to the front way only, an additional machined surface, preferably on the underside of the back way, is necessary to receive the strip which prevents such tilting. This adds to the cost, but the guiding on the front way only enables the saddle to move in closer contact with its guide—and so reduces the tendency to cross-wind—than it can do when the saddle spans across two or more ways.

In the latter case the amount of tightening required to produce the same effect would spring the sheers, and so destroy the alignment. It is obvious that the amount of cross-wind which is possible in a saddle of given length depends upon the closeness of its fit upon the bed. If the amount of tightness necessary to prevent bad cross-winding is sufficient to spring the sheers together, it is clear that a certain amount of binding of the saddle cannot be prevented when it spans more than one way. It is possible, however, by improving the design of section of bed, to avoid this improper springing, and yet to retain the saddle guiding ways on the outer edges of the two outside sheers. This is explained below.

Cross-wind.—The amount of cross-wind which can accompany a given easiness of fit of the saddle upon its guiding edge diminishes as its length increases, and it is only on the ground that the saddle can be made longer when it is guided on the front way only that the narrow guide is to be preferred. With a serious amount of cross-wind, the contact between saddle and bed-edges is restricted to so small a space next the two opposite corners that the oil film is squeezed out, and the frictional resistance to the sliding feed motion is largely increased.

The amount of cross-wind, or the value of the angle which the edge of the saddle makes with the edge of the bed, does not depend, as is frequently stated, upon the ratio of length to breadth of the saddle, but only upon its *length*. This may be proved as follows (Fig. 231):—

Let a saddle be of length l and breadth b, and let the slack be dx. When cross-wind takes place, K moves to N, and the angle of cross-wind is obviously KMN = a, say. Now, the circular measure of a is $\frac{\text{KN}}{\text{KM}}$; and since a is very small, KN is at right angles to MK, and the triangle KON is similar to KAM.

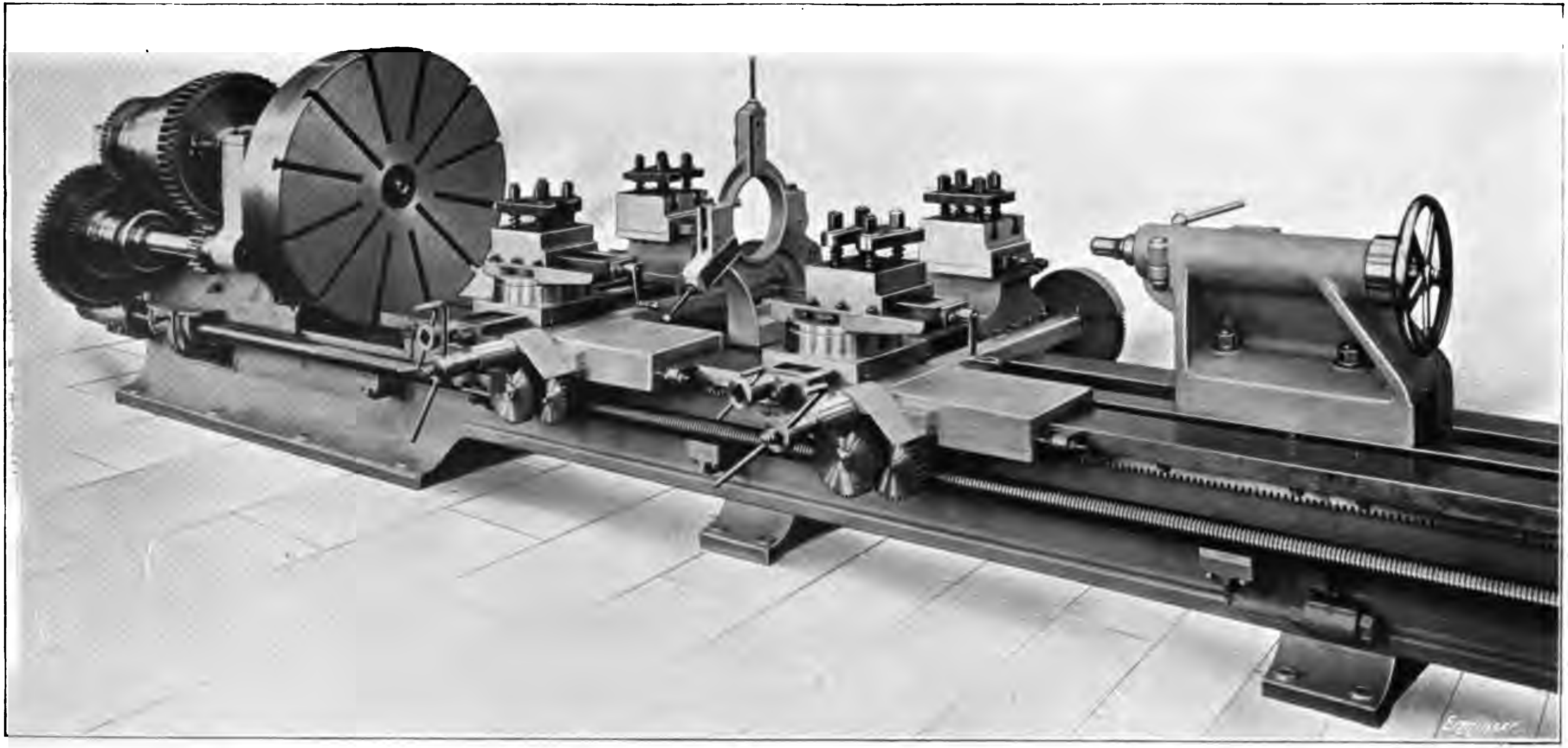


FIG. 165.—18-INCH CENTRE TREBLE-GEARED LATHE (RYDER).

$$\therefore \frac{KN}{KO} = \frac{KM}{KA}$$

$$\therefore a = \frac{KN}{KM} = \frac{KO}{KA} = \frac{dx}{l}$$

or $$a = \frac{\text{slack}}{\text{length}}$$

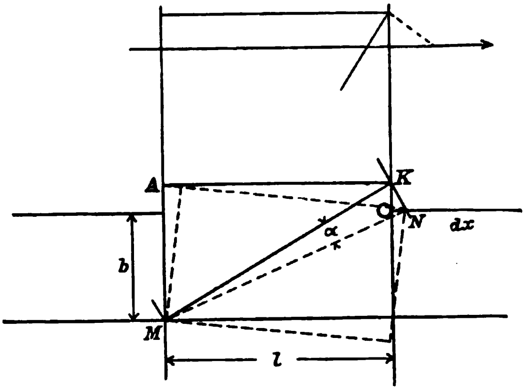


FIG. 231.

Thus it is clear that to prevent too highly localized a pressure in the oil film between the edges of bed and saddle—in other words, to prevent undue cross-wind, it is necessary to make it long; but nothing is gained, *from this point of view*, by increasing the ratio of length to width *merely*. The length itself must be increased in order to secure a diminution of the angle of cross-wind which will accompany a given amount of slack.

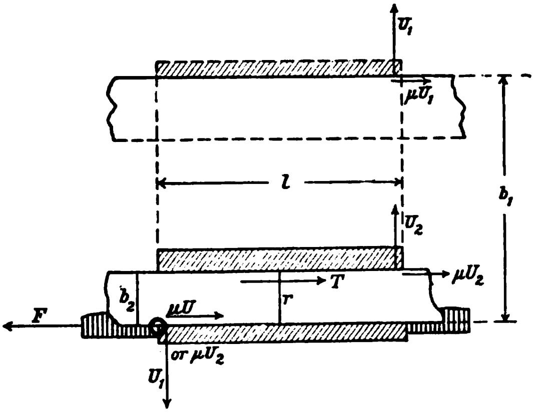


FIG. 232.

It has been claimed by makers of the narrow-way type of bed that a great diminution of the power required to drive the saddle is effected by reason of the reduction of the moment of the frictional resistances caused by the pressures against the guiding edges which this type produces.

There is a small reduction; but that it is so small as to be practically negligible can easily be shown as follows:—

Fig. 232 shows the two types of saddle in plan—the narrow type full, the wide dotted; and the forces shown are those with which the bed acts upon the saddle. Assuming for simplicity that the centre line of the rack coincides with the vertical plane of the front edge of the bed—the driving force acting in that plane being F_1 or F_2—and taking moments about O, we have for—

The wide type—

$$Tr + \mu U_1 b_1 = U_1 l$$

$$\therefore U_1 = \frac{Tr}{l - \mu b_1}$$

The narrow type—

$$Tr + \mu U_2 b_2 = U_2 l$$

$$U_2 = \frac{Tr}{l - \mu b_2}$$

Assume $b_1 = l$, $b_2 = \frac{1}{10}l$, $r = b_2$, and $\mu = \frac{1}{8}$,
Then, for the wider type—

$$U_1 = \frac{b_2}{l - \mu b_1}T = \frac{\frac{1}{10}}{1 - \frac{1}{80}}T = \frac{8}{70}T$$

Also $$F_1 = T + 2\mu U_1 = T + \frac{1}{4} \times \frac{8}{70}T = 1{\cdot}286T$$

For the narrow type—

$$U_2 = \frac{b_2}{l - \mu b_2}T = \frac{\frac{1}{10}}{1 - \frac{1}{8}}T = \frac{8}{79}T$$

$$F_2 = T + 2\mu U_2 = T + \frac{1}{4} \times \frac{8}{79}T = 1{\cdot}253T$$

Thus F_2 is about 2½ per cent. less than F_1; or the driving force for the narrow way is only 2½ per cent. less than for the wide-way type. We see, then, that the narrow-way type of bed retains but one single material advantage over the wide-way type of the same length—that, viz., it obviates the tendency with the latter of the sheers to spring together, which is caused by the tightening up of the taper strips on the edges.

When it is remembered that this springing, which is usually attendant upon the wide-way type as ordinarily designed, may be entirely avoided by using a properly shaped section of bed; that an extra facing has to be provided at the back of the bed to prevent the saddle from tilting with a narrow-way guide; and that a side spring of the top flange of the front bed-girder due to the backward thrust of the tool is difficult to avoid with the latter type, owing to its being naturally weak for resisting such twisting, the utility of the narrow-guide type of bed as against the wide-guide type, when properly designed and proportioned, may be seriously called in question; and, indeed, the balance of advantage, whether for economy, accuracy, or durability, will be found by the impartial investigator to be entirely against it.

Wear of Edge-guides.—In how many ways soever the saddle-guiding surface may have been constructed, their designs seem without exception to err in one respect. *The area of the edge surfaces*—whether inclined or vertical—*has been made far too small.*

Our lathe dynamometer experiments have shown that the backward thrust of the saddle against its edge guide is about half of the downward force which presses the saddle upon the top of the bed—or that the surfacing force is one-half of the vertical force. It therefore follows that the edge-guide area should be half of the top-guide area. This is not, and never has been, carried out in practice. Either, then, the top-guide area is too large, or the edge-guide area is too small.

The large area usually provided for the top guides has no doubt been due to the great flat tops of the I-section girders forming the bed, which were available; but, even if the top-guide surface has been greater than was necessary to prevent wear, it seems clear that the area provided for edge-guiding has been universally made too small. Hence the need for tapered strips and other arrangements for taking up edge wear.

If a suitably large area can be provided on the edge guide—as, *e.g.*, in the manner suggested below—*there seems no reason why provision for adjustment should be necessary for the edge guides any more than it is for the top guides!* It would, however, be essential that the overhanging knee of the saddle should be thoroughly rigid, otherwise the extra surface for edge guiding would not really be available.

Inverted Vee Guide Ways.—Much has been written concerning the relative merits of the inverted vee and the flat top designs for the guiding surfaces of lathe beds. The former is an early type of bed discarded in this country and often called the "American bed." Some of the leading continental makers employ the former type for small lathes, and the latter type for the larger sizes. Others, again, adopt a combination of both, or use flat surfaces only. With the exception of Wm. Sellers and Co. (see Fig. 160, *ante*) the inverted vee is as exclusively used in America as the flat way is in Britain.

Before discussing the subject it may be well briefly to state the principal arguments advanced for and against each type.

The advantages claimed for the inverted vee are as follows:—

(*a*) The vertically downward force on the tool due to the cut, together with the weight of the saddle and rests, hold the saddle to the ways and compel it to follow the same. The saddle is thus provided with an automatic adjustment; cross-winding is prevented, and side strips are rendered unnecessary.

(*b*) Any irregularity or inequality of wear throughout the length of the bed does not affect the force necessary to move the saddle or the uniformity of motion of the same owing to its being free to adapt itself to the bed, and the saddle is therefore much easier to move.

(*c*) The diameter of the work is not appreciably affected by the inequality of wear in the length of the bed, as such wear takes place mostly in a vertically downward direction.

(*d*) The inclination of the surfaces renders them less liable to injury by chips, metal particles, resting of tool, or being used as an anvil.

(*e*) The operator cannot sit on it.

In considering these supposed advantages, it must be premised that it is quite impossible for all four of the vee surfaces of the saddle to fit closely upon their mates upon the bed. Even if they are made to do so by careful scraping when new, the unequal temperatures assumed by bed and saddle when the lathe is at work will immediately destroy the uniformity of fit. By leaving the inside surfaces of two vees clear of each other, the elongation of the saddle when—as usual—it becomes warmer than the bed will no doubt be compensated for by the dropping of the saddle so as to remain tight upon the two outside surfaces. But in that case the insides of the vees serve no useful purpose. This action, whether taking place as the result of temperature variation, or of actual wear of the surfaces, is, no doubt, what is referred to as an "automatic adjustment, rendering side strips unnecessary."

An adjustment for wear, whether automatic or not, is indeed highly necessary with the design of inverted vee guiding surfaces. It is easy to prove mathematically that wear must take place at about *ten to twelve times the rate* with the inverted vee guides that it will with flat top guides.

It will therefore be very necessary that—according to argument (*b*)—the vee type of saddle should be "free to adapt itself to its bed"!

The wear due to the backward push of the tool will also be at least as great in the case of inverted vees as for the inclined edge guides of the flat top design, so that the advantage obtained by reason of argument (*c*) is by no means obvious.

With regard to the advantages claimed under (*d*), it may be remarked that, whatever the type of bed may be, there is always a liability to scoring due to chips falling on the bed and getting between the same and the saddle. But fitting felt pads on the saddle (Fig. 157, *ante*) so that one may clear away the cutting whilst another immediately behind oils the way, will satisfactorily overcome the difficulty. These pads should be easily removed and replaced for cleaning purposes, and should be fitted with a spring so that they may be made to bear continually on the bed. Devices other than this have been tried, but not with the same degree of success or of general application.

The only valid argument in favour of inverted vees is, in our opinion, that advanced under (*e*), viz. *that the operator cannot sit upon them!* It occurs to us, however, to suggest that this might be obviated in the case of flat top guides by supplying wooden boards armed with Swedish nails inverted, and insisting upon their being kept upon the flat bed.

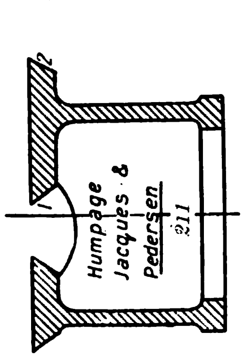
Taken all in all, inverted vees appear to us to be one of the crudest and most imperfect devices yet applied to machine tools. Presumably they are cheaper to manufacture than a properly designed bed with flat top-ways and inclined edge-ways fitted with strips; and if this be so it is the only recommendation they possess.

They ought never to be used for any but the lightest and cheapest lathes, such as brass finishers', wood turners', or such-like machines.

To substantiate this indictment it is, indeed, only necessary to state the case in favour of the flat top bed. Its advantages over the inverted vee type are—

(1) It has an abundance of bearing surface, and the wear is consequently so small that it need not be trued up once in ten years of hard service. Since, therefore, no irregularities are produced by wear, there is no need of "automatic adjustment of the saddle to bed."

(2) The saddle is well supported all across the bed. It may, therefore, be thinner than the vee type saddle for a given rigidity, and will therefore allow



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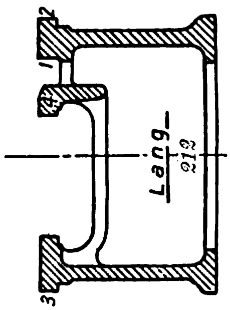

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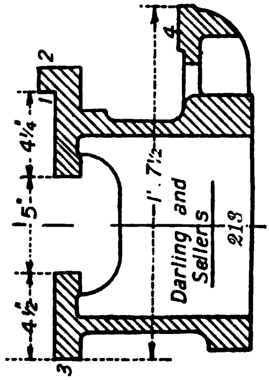

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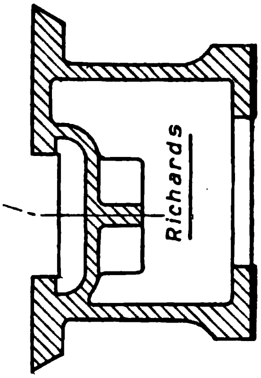

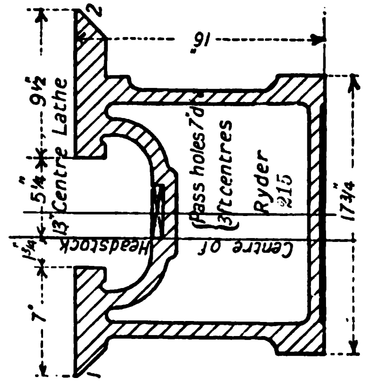

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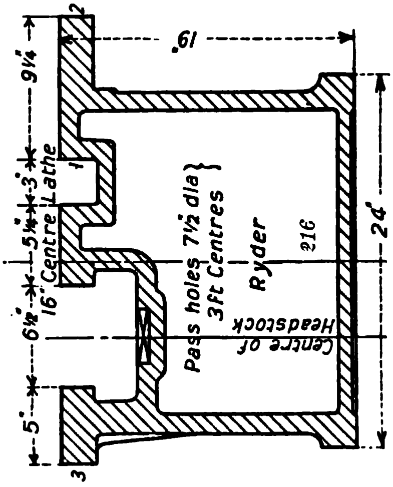

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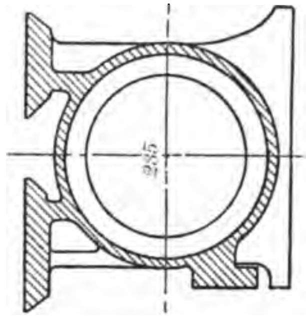

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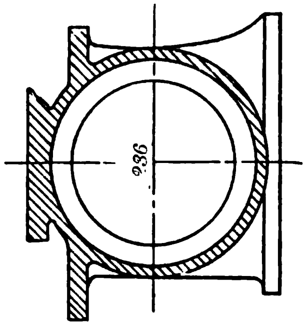


236

of a larger diameter of "swing." The latter type of saddle is inherently bad from the point of view of stiffness.

(3) The side wear is less upon the inclined edge of the flat top design than upon the inverted vees, because, first, the available area to resist wear is twice as great for the former as for the latter; and, secondly, the inclined edge is not subject to the wear due to the vertical forces, as the inverted vees are.

(4) The oil is less liable to be squeezed out of the flat guides than out of the vees, because the pressure per square inch is not nearly so great in the former case.

As we have already pointed out, lighter cuts are generally called for in American practice than in British; and these light cuts are in America taken with a fine traverse, and consequently at a high cutting speed. It need scarcely be pointed out that, since the publication of the Manchester Committee's Report, it has become common knowledge that such light cuts are not economical. More metal can be removed in a given time with moderate or heavy than with fine cuts, and the friction work lost is a larger proportion of the whole, the lighter the cut and the higher the spindle speed. [Provided both the moderate and light cuts are comparable, as being taken through the same back gear.]

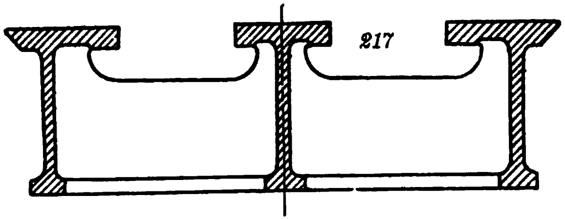
It would almost appear as if the American practice of taking very light cuts at high speeds is the direct result of badly designed beds, which could not suffer the greater forces which are operative with the moderately heavy cut which must be taken for true economy of power.

Typical Bed Sections.—A collection of typical sections of bed as designed for use in small, medium, and large sizes of lathes has been prepared, and is reproduced in Figs. 208 and 236.

Nos. 208, 211–216 represent the sections most commonly used for lathes from 6-in. to 18-in. centres. Between 18-in. and 30-in. centres a bed illustrated by No. 209 is often employed, whilst for lathes between 30-in. and 48-in. centres a double girder section of quasi-box form is frequently adopted. In these large machines, or from 30-in. centres upward, the bed may have two, three, four, or even five ways, as previously mentioned, and the sections in general use for such are shown at Nos. 217–223.

As these beds frequently exceed 10 ft. in width—in the 108-in. centre lathe for steam turbine rotors, by Hulse and Co., illustrated in *The Engineer*, June 1, 1906, the bed was 18 ft. wide—and are beyond the capacity of the available planing machine, or inconvenient to cast or ship, they are made in two or more cross sections, bolted and keyed together (No. 223). In the continental design by Ducommon, shown in No. 222, two independent beds are used and bolted to common base-plates.

Dealing first with small beds, in the ordinary design shown in No. 208, the saddle is supported on the two upper surfaces and guided by the two outside inclined surfaces (1) and (2). The loose headstock fits between the ways and rests on the inner portion of the top surfaces of which the saddle is clear. In the design shown in No. 211 the saddle is supported by the front way only, and guided by the inclined faces (1) and (2) of the same. Similarly with the loose headstock on the back way. In No. 224 the saddle fits on the side surfaces at the front of the bed and the loose headstock on the upper horizontal ways. The arrangement of the ways in Nos. 211, 224, and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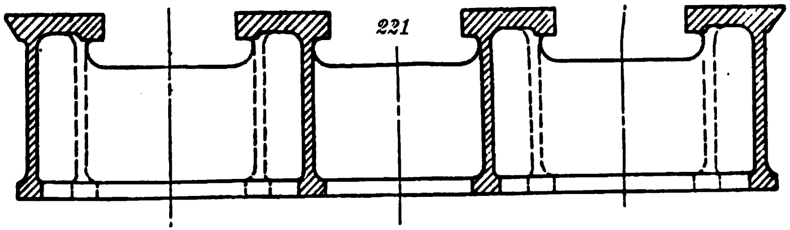
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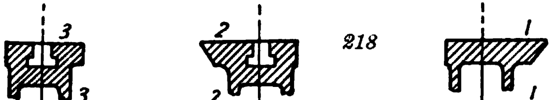
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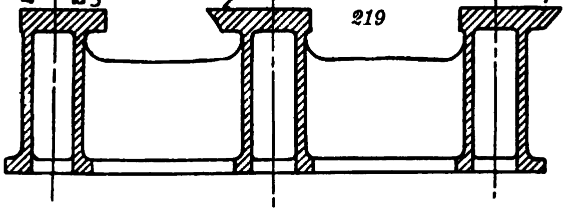
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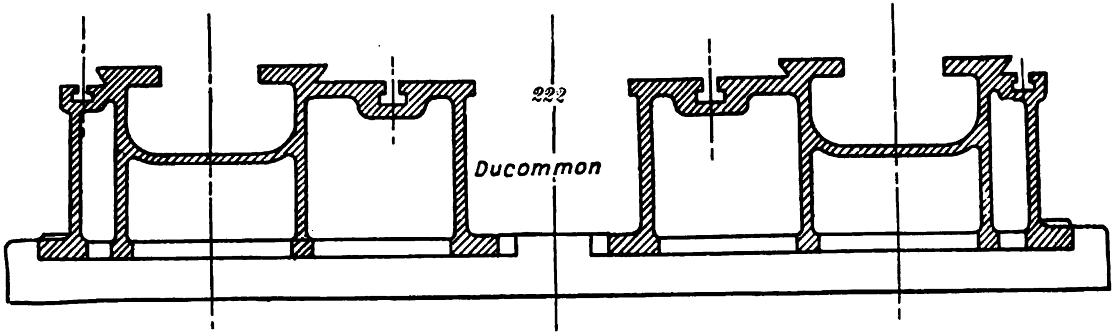
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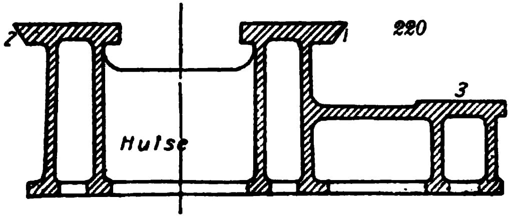
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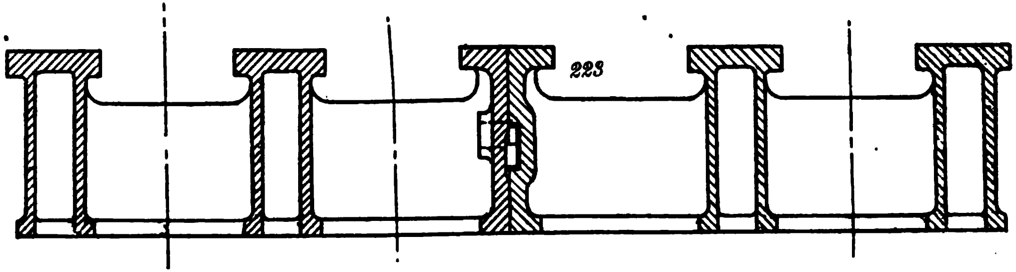

223

225 is such that the saddle can be moved beyond the loose headstock without disturbing the same.

The sliding head of No. 212 is supported on the back way and a separate inner way (4), which is inclined on the outer top side, so that as the head is bolted down it is pulled hard against the left-hand vertical face of (4), thus automatically adjusting or aligning the same. The saddle bears on the outer ways, but is guided by the front-way vertical faces (1) and (2), and is clear of the surface at (3). A strip fitted to the saddle bears on the horizontal under face of (3), and prevents it from tilting (see Fig. 155, *ante*).

Similar guiding and tilt prevention means to the saddle are employed in the designs shown in Nos. 213, 216, 226, and 227. In addition to the upper ways (No. 213), a support to the saddle is furnished at (4), after the manner of that shown in No. 220. This considerably reduces the overhang of the saddle without inconvenience to the operator. It also allows of the actuating mechanism to the saddle being placed between the upper and lower front ways, and under the average position of the tool, so that the tendency to cross-wind is reduced.

The design shown in No. 227 is by Mr. W. P. Hunt in the *American Machinist*. It is intended to overcome the trouble occasioned by metal particles falling on the ways. The flat surface whereon the saddle slides is situated well forward, and an independent surface inclined at 30 deg. to the horizontal is provided to receive the loose headstock and divert the cuttings into the pan at the rear and clear of the operator's feet. The faces at (3) are protected by the projecting ledge of the inclined surface, whilst the main way near to the tool is guarded by the long wing of the saddle.

The American beds are usually provided with two pair of vees, an outer pair upon which the carriage slides (1), (1), No. 228, and an inner pair to receive the sliding headstock (2), (2), No. 228. Occasionally the inside pair of ways is dispensed with, and the headstock made to slide on the flat surfaces. The provision of separate surfaces is essential if parallel work is desired, since then the wear produced by the carriage in sliding along the ways does not affect the height of the loose headstock centre. In order to secure a larger bearing surface for the saddle, the Lodge and Shipley Company has displaced the inner front vee by a flat surface, upon which the carriage is bedded, whilst retaining the outer vees as guiding surfaces to the same (Fig. 157, *ante*). The effect of this is to reduce the span of the saddle and provide a support to it immediately under the tool, but the saddle no longer "automatically adjusts itself" to the ways. Another recent design by the same firm is shown at No. 225, where the loose head is carried on the inclined back surfaces, whilst the horizontal flat surfaces are prepared to receive the saddle. This is, therefore, a return to the British type of bed in the latter respect. The design shown at No. 226 is a recent patent by Mr. J. Brockie, who claims an improved relation of the surfaces to one another, also an improved means of adjustment. The front and back upper surfaces have parallel inclinations at about 30 deg. to the horizontal, whilst the saddle is guided by the vertical face (1) and the inclined face (2). The strips at (2) and (3) are intended to prevent tilting. They are adjusted by pins which are concentric in the saddle and excentric in the strips. The loose headstock is supported on the horizontal central flat ways. In this design the objection to the inverted vee ways that the saddle can be forced up the incline is obviated; the horizontal thrust of the tool being taken upon the vertical surface at (1).

No. 229 shows a section for a universal lathe whereon a sleeve circular at its periphery embraces the bed and is propelled along the same by a screw in the centre of the gap and directly below the top surface. The saddle is mounted and free to swivel on the sleeve, whilst the loose headstock rests upon the bed direct.

The design of bed shown at 214 is intermediate between the girder and box types, as shown at 208 and 215. A continuous plate runs the full length of the bed between the ways, having pass holes only at intervals with deep return flanges to compensate for the metal removed. The bottom flanges are similar to that of the ordinary lathe bed. The saddle is supported by the two top faces, and guided by the outer inclined side surfaces. Owing to the sheers being connected by the metal plate, the ways cannot be sprung together when tightening the side strips, as can be done in the ordinary design. The rigidity due to the full box type of section is, however, not attained.

Such box beds are shown by Nos. 215 and 216; and their power to resist torsion and bending as compared with the ordinary designs has been worked out below. Their introduction by Messrs. Ryder was the result of a suggestion by the authors. Any one may convince himself of the superiority of sections 215 and 216 over 214 by a simple experiment suggested by Capt. Sankey. If a cigar-box be taken with the lid off, it will be found that it can be quite easily twisted out of shape; but if the lid be closed, the full strength of the hands is unable to effect any distortion.

In No. 215 the ordinary supporting and guiding surfaces are employed, whereas in No. 216 the saddle is supported on the three horizontal ways and guided by the vertical faces (1) and (2), whilst tilting is prevented by a strip bearing on the underside of (3). It will be observed that the centre line of the headstock and bed are not coincident, thereby affording a greater bearing surface to the saddle at the front and reducing the overhang of the same. In machines fitted with front rests only this practice is frequently adopted.

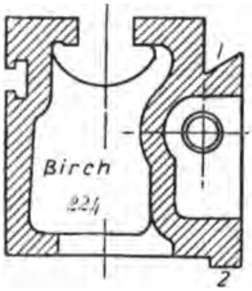
In lathes designed for light crank-shaft work the front and back rests may be mounted on independent saddles. As these are made to clear each other, the front saddle is supported on the front and centre ways—in the case of the three-way bed—but is guided by the front way only. Similarly with the back saddle on the back and centre ways (No. 217). Usually, however, in this class of machine, or in lathes over 30-in. centres, a four- or five-way bed is adopted (Nos. 221, 222, and 223).

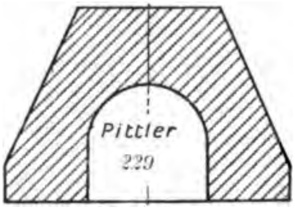
The tee slots in the upper surfaces of Nos. 218 and 222 are to receive the loose headstock holding-down bolts, whereas in the other cases clamping-plates are employed to hold the head fast to the bed, and fit on the underside of the projecting lips as in the smaller machines.

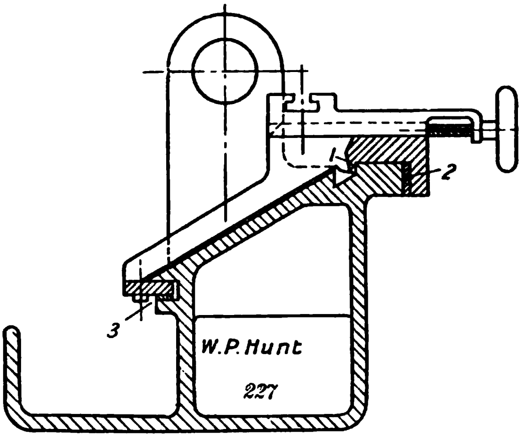
In the three-way pattern, shown at No. 220, the carriages bear on the three upper surfaces (1), (2), and (3), but are guided by the two inclined faces of (1) and (2). The front way (3) is dropped to allow of the saddle being deepened to give rigidity when the tool and rests are in operation in their extreme outward position. In the 108-in. centre lathe, previously referred to, the saddle spans the four ways, but is guided by the two centre ways only.

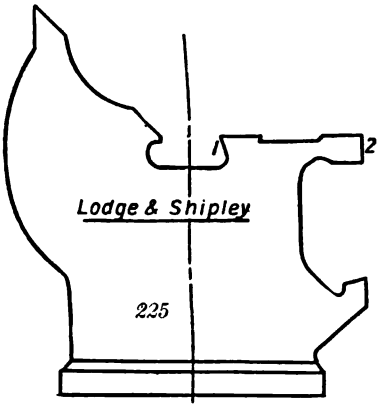
The sections of these beds are usually of the double girder pattern for each sheer, although a three- and four-way single-web pattern, as shown in full lines at 217 and 221, are occasionally used. The full and dotted lines of the latter figure represents the section most commonly adopted for four-way beds.

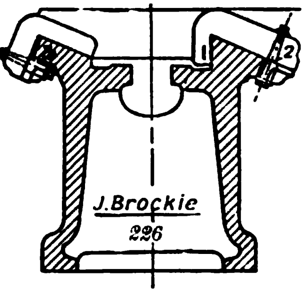
Strength and Stiffness of Different Sections of Bed.—The causes producing the principal deflections in a lathe bed are two in number.

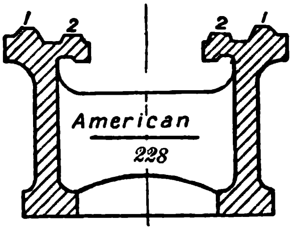












First, the force exerted when a heavy cut is taken by a single tool acting, say, midway between the centres, which tends to bend the job upwards and the lathe bed downwards. It also tends, of course, to twist the job relatively to the face-plate, and to oppositely twist the bed relatively to the fast head-stock; this action being the more severe, the greater the diameter of the work for a given cut.

In such a case the lathe bed may be regarded (neglecting the twist) as a girder loaded in the centre and supported at the ends. It is to a slight extent encastré, the moment of encastrément being $\frac{1}{8}Fx$, where F is the force due to the cut, and x is the distance measured along the lathe axis between the end of the job and (the vertical plane containing) the holding-down bolts of the headstock which are nearest to the job. Neglecting this, the bending moment for the standard cut $\left(\text{area } \frac{h^2}{6400}\right)$ will be $\frac{1}{4}Fl$; where l is the distance, measured along the bed, between the pair of holding-down bolts of the two headstocks which are nearest to each other. This distance may be taken at $8h$. Thus the bending moment at the centre (that is, the maximum to which the bed is liable) when the standard cut is being taken is approximately equal to—

$$\begin{aligned} M &= \frac{2240 \times 100h^2}{6400} \quad \frac{8h}{4} \\ &= 35h^2 \times 2h \\ M &= 70h^3 \text{ lb.-ins.} \qquad (151) \end{aligned}$$

(Here the cutting pressure has been taken equal to 100 tons.)

Secondly, the force exerted when a heavy cut is taken by a single tool operating on work of face-plate diameter anywhere along the bed, which tends to twist the lathe bed relatively to the fast headstock. The twisting moment (T) is Fh; and, with $F = 35h^2$ as before, we have—

$$T = 35h^2 \times h = 35h^3 \text{ lb.-inches.}$$

With two tools, cutting on opposite sides of work of face-plate diameter as above, the cut would probably be of about half the standard area for each; and the twisting moment would then be the same as before, viz.—

$$T = \frac{1}{2} \times 35h^2 \times 2h = 35h^3 \text{ lb.-inches.}$$

(In this latter case there would be practically no bending moment.)

Thus the maximum twisting moment to which the bed is liable for the standard cut taken upon face-plate diameter is—

$$T = 35h^3 \qquad (152)$$

The angle of twist of the bed will obviously be greater the further away the tool is acting from the fast headstock. If the greatest distance therefrom at which it can operate be taken equal to $8h$ (taking the length equal to the span formerly assumed for bending), we shall be able to compare the two deflections, due to bending and twisting respectively—qualitatively, at all events, if not quantitatively—and to determine which of them is the more important from the point of view of rigidity or maintenance of alignment.

It is found, upon examination, that the torsional effect is the more serious, and we shall proceed to examine this first.

The formula which gives the relation between the twisting moment, the area (a), the modulus of rigidity (G), and the angle of twist in radians per unit length (i_0), *for a shaft of circular section*, is well known to be—

$$T = JGi_0 \qquad (153)$$

where J is the polar moment of inertia of the circular area of radius r; J being equal to $a\frac{r^2}{2}$ or $\frac{a^2}{2\pi}$.

Hence
$$T = \frac{a^2}{2\pi}Gi_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (154)$$

If the section be of any other form, *e.g.* square, rectangular, or of any of the shapes usually employed for lathe beds, it may be easy to determine J; but it is found that formula (3) is no longer applicable, being, as was shown by de Saint Venant in his great memoir on "Torsion" (see also Todhunter and Pearson's "History of Elasticity," Vol. II.), in that case not even approximately true.

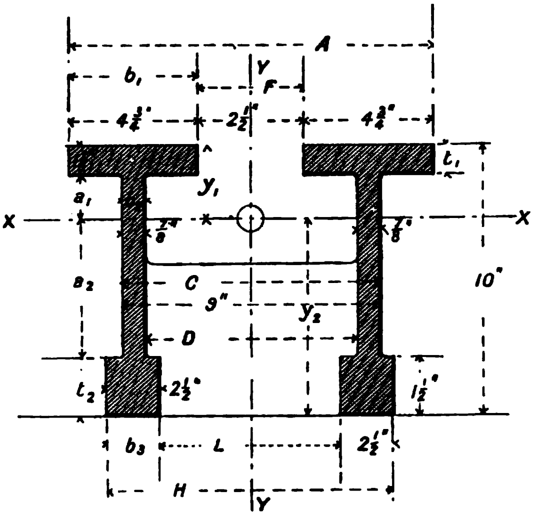


FIG. 233.

As a practically useful result of his researches, he proposed the formula—

$$T = \frac{1}{40}\,\frac{a^4}{J}Gi_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (155)$$

as applicable to every shape of section, whether round, square, I-shaped, solid, or hollow. The practical accuracy of this formula for angles of twist within the working strengths of the material was established by Bauschinger.

It is easy to show that this expression gives a nearly correct result for a circular—solid or hollow—section. Thus $T = \frac{a^4}{40J}Gi_0 = \frac{a^4}{40ar^2/_2}Gi_0 = \frac{a^2\pi}{20}Gi_0$; but by the ordinary formula $T = \frac{a^2}{2\pi}Gi_0$; and the former value is equal to the latter if $2\pi = \frac{20}{\pi}$; that is, if 6·2832 = 6·38, which is nearly enough true.

We may now show how to calculate for a lathe bed of given section and polar moment of inertia J; which is of cast-iron ($G = 6{\cdot}6 \times 10^6$ lb. per sq. in.); of area a square inches; and which is subjected to a given twisting moment T lb.-inches, the angle of twist i_0 due to a given twisting moment of, say, $T = 35h^3$ lb.-inches; and hence to determine the deflection at the extreme outer edge of the bed, h inches from the centre and at a distance $8h$ from the fast headstock.

We take the following as a typical section for a 6-in. bed:—

If Ixx be the moment of inertia of the section about the axis XX, and Iyy the moment about the axis YY, then J, the polar moment of inertia of the section, or that about an axis through the intersection of XX and YY, and perpendicular to their plane, is known from—

$$\mathrm{J} = \mathrm{I}xx + \mathrm{I}yy.$$

To find Ixx, we must first determine the position of the centre of gravity of the area of the section; then, referring to the figure, the value of Ixx will be got from—

$$\mathrm{I}xx = \tfrac{1}{3}[b_1(y_1{}^3 - a_1{}^3) + b_2(a_1{}^3 + a_2{}^3) + b_3(y_2{}^3 - a_2{}^3)]$$

Similarly, Iyy will be obtained from—

$$\mathrm{I}yy = \tfrac{1}{12}[t_1(\mathrm{A}^3 - \mathrm{F}^3) + (a_1 + a_2)(\mathrm{C}^3 - \mathrm{D}^3) + t_2(\mathrm{H}^3 - \mathrm{L}^3)]$$

Thus J is known.

The values of J have been calculated in this way for the three two-way beds whose dimensions are given above, and are tabulated herewith—

h	in. =	6	12	18
a	sq. in. =	31·1	63·6	112
y_2	in. =	5·57	8·5	10·68
Ixx	in.^{4} =	408	1484	4,800
Iyy	" =	444	3613	15,650
J	" =	852	5097	20,450
$\frac{a^4}{\mathrm{J}}$	" =	1095	2900	7,720

We may now proceed to calculate the deflections due to (a) torsion, (b) bending, by the torsional and bending moments respectively specified above.

(a) Torsional deflection at a point in the edge of the bed distant $8h$ in. from the fast headstock, the edge being distant (practically) h in. from the centre of gravity of the section, and the sections being of the form and dimensions figured above. If δ be this deflection (in inches) and i the corresponding angle of twist (in radians), then $\delta = ri\ (= l\phi)$; also $i = i_0 l$.

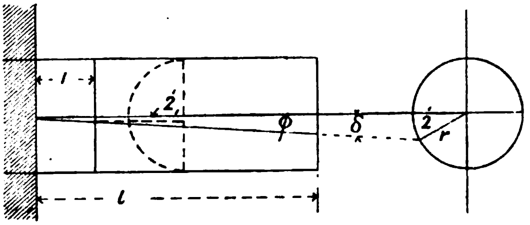


FIG. 234.

$$\therefore \delta = rli_0$$

but $$i_0 = \frac{40\mathrm{JT}}{\mathrm{G}a^4}$$

$$\therefore i_0 = \frac{40\mathrm{JT}}{\mathrm{G}a^4}rl$$

With $r = h$ and $l = 8h$, $rl = 8h^2$; also $T = 35h^3$

Therefore
$$\delta = \frac{1\cdot7}{10^3}\;\frac{h^5}{\dfrac{a^4}{J}}$$

This gives—

$$\begin{array}{lccc}
\text{For } h = & 6'' & 12'' & 18'' \\
\delta = & 0\cdot0147 & 0\cdot1175 & 0\cdot397 \text{ ins., if } \dfrac{a^4}{J} = 25h^2
\end{array}$$

$$\begin{array}{lccc}
\text{Actual value is } \delta = & \dfrac{1\cdot7 \times 6^5}{10^3 \times 1095} & \dfrac{1\cdot7 \times 12^5}{10^3 \times 2900} & \dfrac{1\cdot7 \times 18^5}{10^3 \times 7720} \\
= & 0\cdot0121'' & 0\cdot1452'' & 0\cdot4162''
\end{array}$$

$$\left.\begin{cases}
\delta = \dfrac{h^3}{15,000} = & 0\cdot014 \qquad 0\cdot10 \qquad 0\cdot39 \\
\delta = \dfrac{h^3}{14,000} = & 0\cdot0155 \qquad 0\cdot123 \qquad 0\cdot416
\end{cases}\right\} \text{approximately}$$

It will be found on trial that this calculated deflection increases at a somewhat greater rate than the square of the height of centres. The reason for this is evident when we observe that the deflection produced by the standard cut at the point in question increases as the fifth power of the height of centres, and diminishes as $\frac{a^4}{J}$ increases. This latter, for the beds as made in practice, only increases at a rate somewhere between the square and the cube of the height of centres.

To obtain the same deflection in all sizes $\frac{a^4}{J}$ should increase as h^5.

(*b*) The deflection due to bending, produced by a tool taking the standard cut, midway between centres, is: if the beam be assumed supported (and not *encastré*) at the ends; and if it be granted that both I-beams of the bed take part equally in resisting the bending action (a supposition which is too favourable to the ordinary bed; the two sheers or girders being only connected together at intervals by shallow cross girders)—given by the expression—

$$\delta = \frac{Wl^3}{48EI_{xx}}$$

Here $W = 35h^2$, $l = 8h$, $E = 17 \times 10^6$, and I_{xx} has the values tabulated above, viz.—

For	$h =$ 6 in.	12 in.	18 in.
	$I_{xx} = 408$	1484	4800
and	$\delta =$ 0·00042 in.	0·00368 in.	8·008 in.

Thus this form of bed is much stiffer as regards bending than as regards torsion.

Beds of Circular Section.—If we take the same amount of metal as is put into the ordinary beds, but make the section a hollow circular one of diameter equal to one and a half times the height of centres, we obtain the following deflections for the standard cut measured at the same point, viz. $8h$ in. from the fast headstock, and h in. from the centre of the lathe.

Deflection due to Torsion.—The formula for the angle of twist per unit length of bed becomes—

$$i_0 = \frac{2\pi \mathrm{T}}{\mathrm{G}a^2} \left(\text{or } \frac{40\mathrm{JT}}{\mathrm{G}a^4} \text{ approximately}\right)$$

and for

$$\delta = rli_0 = \frac{2\pi \mathrm{T}}{\mathrm{G}a^2} rl = \frac{2\pi \mathrm{T}}{\mathrm{G}a^2} 8h^2$$

or

$$\delta = \frac{16\pi \times 35h^5}{6{\cdot}6 \times 10^6 a^2} = \frac{266}{10^6} \quad \frac{h^5}{a^2}$$

For h =	6	12	18
h^5 =	7780	248,900	1,890,000
a^2 =	$31{\cdot}1^2$	$63{\cdot}6^2$	112^2
=	965·8	4,042	13,534
$\delta = \frac{2{\cdot}66h^5}{10^4 a^2}$ =	0·00214	0·0157	0·040
δ =	0·002 in.	0·0157 in.	0·040 in.
Stiffness relatively to bed of ordinary design	6 times	9 times	10 times as stiff

The beds of circular section illustrated by Figs., Nos. 235 and 236, p. 325, which have about the same moment of resistance to bending as the ordinary form of bed, will have about the same deflection in transverse bending as the latter.

CONCLUSIONS.

The conclusions which must be drawn from this investigation are the following:—

(1) Lathe beds of the ordinary design, consisting of two or more girders laid side by side, connected only at intervals by relatively shallow cross beams, are very weak to resist the torsional moments to which they are subjected, and which tend to produce much greater strains and deflections in them than the bending moments do.

This weakness increases enormously as the size of the lathe increases, so that, were it not for the additional stiffness provided by the great masses of concrete which they rest upon and are securely connected to, the beds would be utterly inadequate to withstand the loads to which they are subjected in large lathes.

(2) Ordinary lathe beds are not seriously strained by the bending moments to which they are subjected: or, in other words, the beds are much stronger for resisting bending than for torsion. Yet, whilst this is true, large lathe beds of ordinary design are proportionally much weaker than small ones. Thus, for an 18-in. lathe the deflection due to bending under the standard

cutting force is found to be twenty times as great as for a 6-in. lathe. If this deflection may be regarded as a true measure of the stiffness of a bed, it ought, however, to be the same for all sizes of lathe. It is certain that such large deflections will be accompanied by great vibration.

(3) The proper form to give the section of a lathe bed is the box or circular. This has been shown above to give a torsional stiffness from six to ten times (and for larger beds still more), and a bending stiffness at least equal to that of a bed of ordinary design when both contain the same weights of metal. The continuity of the metal between the sheers further obviates the springing due to the tightening of the taper strip in saddles which span the whole bed.

If such a bed section were adopted its strains would be self-contained. No assistance from a strong foundation would be required, and no greater a foundation would be necessary than what would merely support the weight.

CHAPTER XXXIX

THE ECONOMICS OF MACHINING IN THE LATHE, AND THE FACTORS WHICH GOVERN THE TOTAL COST

THE cost of machining off one unit of weight of either steel or cast iron depends on the following factors:—

I. Charge for power: (*a*) useful,
(*b*) frictional.
II. Charge for labour: (*a*) skilled,
(*b*) unskilled.
III. Charge for establishment: (*a*) land, buildings, insurance, rent, etc.
(*b*) management.
IV. Charge for first cost of the Machine Tool: (*a*) interest,
(*b*) depreciation.

I. **Power.**—It has been found that about 2 horse-power must be developed *at the tool* to turn off either steel or cast iron continuously at the rate of 1 lb. per minute. In other words, to remove one pound requires the expenditure of 66,000 ft. lbs. of net work at the cutting edge.

It is further found that the work lost in friction, when a lathe is kept running for one minute, is about 3300 ft. lbs. per inch of height of centres of the lathe, whether the lathe be cutting or running light. In other words, the frictional or waste horse-power of a lathe increases proportionally to its size and varies but little with the load. It may be taken roughly at $\frac{h}{10}$ horse-power (h = height of centres, inches).

It is clear that the greater the proportion of cutting time to running time and the heavier the cut taken when cutting, the smaller fraction will the waste power be of the total required per pound cut.

Thus, if W lbs. can be machined off during 60 minutes' running time of the lathe, the total work done will be $[66{,}000\ W + 60 \times 3300h]$ ft. lbs.

The work done will therefore have been

$$\left(66{,}000 + \frac{198{,}000h}{W}\right) \text{ ft. lbs. per lb. cut.}$$

If the power costs 0·75 penny at the power station, and (owing to frictional loss, etc., *en route*) it costs $1\frac{1}{2}d.$ at the lathe per horsepower-hour, then the charge for power (P) for the above pound cut will be (since 1 hp.-hr. = 1,980,000 ft. lbs.)—

$$\left(\frac{66{,}000}{1{,}980{,}000} + \frac{198{,}000h}{1{,}980{,}000\ \mathrm{W}}\right) 1{\cdot}5 \text{ pence per lb.}$$

or $$\mathrm{P} = \left(\frac{1}{20} + \frac{h}{6\frac{2}{3}\ \mathrm{W}}\right) = 0{\cdot}05 + 0{\cdot}15\,\frac{h}{\mathrm{W}} \text{ pence per lb. cut.}$$

or $$\mathrm{P} = 112 + \frac{336h}{\mathrm{W}} \text{ pence per ton cut} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (156)$$

It must be remembered that this is the charge for power irrespective of the ratio of *running time* to *working time*. Thus, whether the W lbs. are machined off in 60 consecutive minutes of running time, or in six periods of ten minutes each of running time spread over 2, 4, or any number of hours of working time, makes no difference to the above figure. Strictly speaking, an added charge for friction of line shafting, etc., should be made in the latter case; but this may be supposed to be allowed for in the price of 1½ pence above assumed.

The proportion of the total possible working hours during which the lathe stands still does, of course, affect the total cost of cutting off a pound of metal; not on the score of power, but on those of wages, establishment, and on cost. These items are subsequently considered.

We shall denote by w (compared with W) the weight of metal removed on the average during each hour the shop works throughout the year.

To find the value of w as a function of the size of lathe; of the ratio of actual to possible running time over the year; of the ratio of the average area of cut actually taken to the standard area for which the lathe is designed, *the actual cutting speed being always that at which the standard cut should be taken if the best possible use is to be made of the tool steel employed,* we proceed as follows:—

Let h = height of lathe centres in inches.

x = ratio of actual to possible hours per annum during which the lathe runs. (It is here assumed that it never runs except when cutting.)

Y = number of hours the shop works per annum (say 3000).

a_s = standard area of cut for which the lathe is designed. $\left(a_s = \dfrac{h^2}{6400}\right)$.

na_s = average area of cut taken throughout the year. (n being usually less than unity.)

$v_s = \dfrac{1}{a_s} + 15$, the standard cutting speed (ft.) when high speed steel is used.

$v_s = \dfrac{1}{na_s} + 15$, the cutting speed (ft., min.) of which the tool steel is economically capable with the area of cut na_s when high speed steel is used. Then, since the weight of one cubic inch of the steel operated upon is about 0·29 lbs., and, *on the assumption that that average cut is always taken at its corresponding best tool speed* $\left(v_s = \dfrac{1}{na_s} + 15\right)$, we shall have ($\mathrm{W}_p$ = possible weight cut)—

$$\begin{aligned} W_p &= 60 \times na_s\left(\frac{1}{na_s} + 15\right) \times 12 \times 0{\cdot}29 \\ &= 200 + 3200na_s \text{ (approximately)} \quad \dots\dots \quad (157) \\ &= 200 + \frac{nh^2}{2} \text{ (approximately)} \quad \dots\dots \quad (158) \end{aligned}$$

lbs. cut per hour of running time; and—

$$w_p = xW_p = 200x + \frac{xnh^2}{2} \quad \dots\dots \quad (159)$$

lbs. cut per hour of shop working time.

This weight is possible during xY hours per annum, *only if the speed appropriate to the given average cut can always be obtained on the lathe and high speed steel be used.*

Thus, *e.g.*, if the average cut taken be half the standard cut, $n = \frac{1}{2}$; if the lathe runs two-thirds of the total working hours of the shop, $x = \frac{2}{3}$;

then
$$\begin{aligned} w &= 200 \times \tfrac{2}{3} + \frac{2 \times 1 \times h^2}{3 \times 2 \times 2} \\ &= 140 + \frac{h^2}{6} \quad \dots\dots \quad (160) \end{aligned}$$

lbs. cut per shop-hour on the average throughout the year.

As a matter of fact, however, the above required cutting speed cannot be obtained on every diameter of job except in the case of lathes upon which a continuous variation of speed from the highest to the lowest is supplied by means of a separate motor with a suitably arranged rheostat, or of a stepless cone or equivalent mechanical device. In general, the required speed v will fall between two successive available speeds, a lower v_1 and a higher v_2, of which the former will be made use of by the turner as being within the proper compass of the tool steel.

If the required speed v be *on the average* halfway between v_1 and v_2, then the cut will be taken at a speed which bears to the proper speed the ratio—

$$\rho = \frac{2v_1}{v_1 + v_2} \quad \dots\dots \quad (161)$$

now, if r be the geometric ratio—

$$v_2 = rv_1; \quad \therefore \rho = \frac{2v_1}{v_1(1+r)} = \frac{2}{1+r} \quad \dots \quad (162)$$

The weight cut per running hour will, in this way, be reduced from W_p to W; where—

$$W = \frac{2W_p}{1+r} = \frac{2}{1+r}\left[200 + \frac{nh^2}{2}\right] \quad \dots\dots \quad (163)$$

the weight cut per shop hour will similarly be reduced from w_p to w; where—

$$w = xW_p = \frac{2x}{1+r}\left[200 + \frac{nh^2}{2}\right] \quad \dots \quad (164)$$

The cost for power then becomes—

$$P = 0{\cdot}05 + \frac{0{\cdot}15h}{W}$$

$$P = 0{\cdot}05 + \frac{0{\cdot}15h}{\left(200 + \frac{nh^2}{2}\right)\frac{2}{1 + r}}$$

or $$P = 0{\cdot}05 + \frac{0{\cdot}15h(1 + r)}{400 + nh^2} \text{ pence per lb.} \quad . \quad . \quad . \quad . \quad (165)$$

or $$P = 112 + \frac{336h(1 + r)}{400 + nh^2} \text{ pence per ton cost} \quad . \quad . \quad (166)$$

Items II. *and* III.—**The Labour and Establishment** charges may be conveniently taken together, as they both depend on the size of the lathe.

II. (*a*).—Turner's wages will be about 9*d*. per hour for all machines above 12-in. centres, but less for smaller sizes; say about $\frac{h}{3}$ pence per hour for the apprentices or boys who work the smaller tools.

II. (*b*).—One labourer can clear away the cuttings from about 3 high speed tools, and his wages may be taken at 6*d*. an hour. For small lathes one labourer can attend to about 6 lathes. The charge for labouring may therefore be approximated to by charging at the rate of $\frac{h}{6}$ pence per hour, as the large lathes may use 2, 4, or even 6 tools at once.

III. (*a*) *and* III. (*b*).—The charges under these heads are very differently estimated in different shops. We shall assume that the proportion of total oncost chargeable to each machine increases with its size and that it will amount to $\frac{h}{2}$ pence per hour.

The total charge under headings II. and III. will therefore amount to

$$\left.\begin{array}{l} \underset{\text{(turner)}}{9} + \underset{\text{(labourer)}}{\frac{h}{6}} + \underset{\text{(oncost)}}{\frac{h}{2}} \text{ pence per shop-hour for lathes above} \\ \text{and } \frac{h}{3} + \frac{h}{6} + \frac{h}{2} \text{ pence per shop-hour for lathes below} \end{array}\right\} \text{12 in.}$$

Say $9 + 0{\cdot}66h$ pence per hour for large lathes
and h „ „ „ small lathes.

If, on the average throughout the year, w lbs. of metal are machined off per hour of possible working time by a lathe of h inches height of centres, then the cost for labour and establishment will amount to $L' = \frac{h}{w}$ pence per pound for small,

$$\text{and } L'' = \frac{9 + 0{\cdot}66h}{w} \text{ for large lathes} \quad . \quad . \quad . \quad (167)$$

Thus—

$$L = \frac{2240h}{\frac{2}{1+r}\left(200x + \frac{xnh^2}{2}\right)} \text{ or } \frac{2240(9 + 0\cdot66h)}{\frac{2}{1+r}\left(200x + \frac{xnh^2}{2}\right)} \text{ pence per ton} \quad (167\text{A})$$

III. *(a) and (b).*—**The first cost** of lathes of good design and finish, with beds about 12 times the height of centres, in length, is found to vary somewhat as the $\frac{3}{2}$ power of that height. For lathes of usual ranges and numbers of spindle speeds we have found the following expression for the first cost in terms of the height of centres, h, and the value of the geometric ratio of consecutive spindle speeds, r.

$$\text{First cost, } £C = £\left[3\cdot7 + \frac{1}{4(r-1)}\right]h^{\frac{3}{2}} \quad . \quad . \quad . \quad (168)$$

If we take interest and depreciation at together 10 per cent., the charge under this head will be $\frac{C}{10}$ pounds stg. per annum or $\frac{240C}{10Y}$ pence per shop-working hour, where Y = number of hours the factory works per year. Thus, *e.g.*, if the working year consisted of 2400 hours, the charge would be $\frac{240C}{24,000}$ or $\frac{C}{100}$ pence per hour.

As w lbs. are cut per factory hour (on the average), the cost chargeable for interest and depreciation is $I = \frac{240C}{10Yw}$ pence per lb. cut.

In what follows we shall assume that the factory year consists of 3000 working hours; then

$$I = \frac{C}{125w} = \frac{\left[3\cdot7 + \frac{1}{4(r-1)}\right]h^{\frac{3}{2}}}{125w} \text{ pence per lb.}$$

and

$$I = \frac{2240C}{125w} = \frac{17\cdot9C}{w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (169)$$

or

$$I = \frac{17\cdot9\left[3\cdot7 + \frac{1}{4(r-1)}\right]h^{\frac{3}{2}}}{\frac{2}{1+r}\left(200x + \frac{xnh^2}{2}\right)} \text{ pence per ton cut} . \quad (169\text{A})$$

The total cost of removing one ton of steel shavings is accordingly

$$T = \quad P \quad + \quad L \quad + \quad I \quad . \quad . \quad . \quad . \quad . \quad (170)$$

$$= \left[112 + \frac{336h}{W}\right] + \left[\frac{2240h}{w}\right] + \left[\frac{2240C}{125w}\right] \quad . \quad . \quad . \quad . \quad (171)$$

$$= 112 + \frac{336xh + 2240h + 17\cdot9\left[3\cdot7 + \frac{1}{4}(r-1)\right]h^{\frac{3}{2}}}{\left[400 + nh^2\right]\frac{x}{(1+r)}} \quad . \quad (172)$$

pence per ton.

We have made use of this expression, based upon the data given above, to estimate the cost of cutting under various conditions, and for various sizes of lathe. The results are given in Table LXI., and have been plotted as curves of cost on various bases in Figs. 237 to 239.

Fig. 237 shows the effect upon the cost of cutting (a ton of mild steel), of

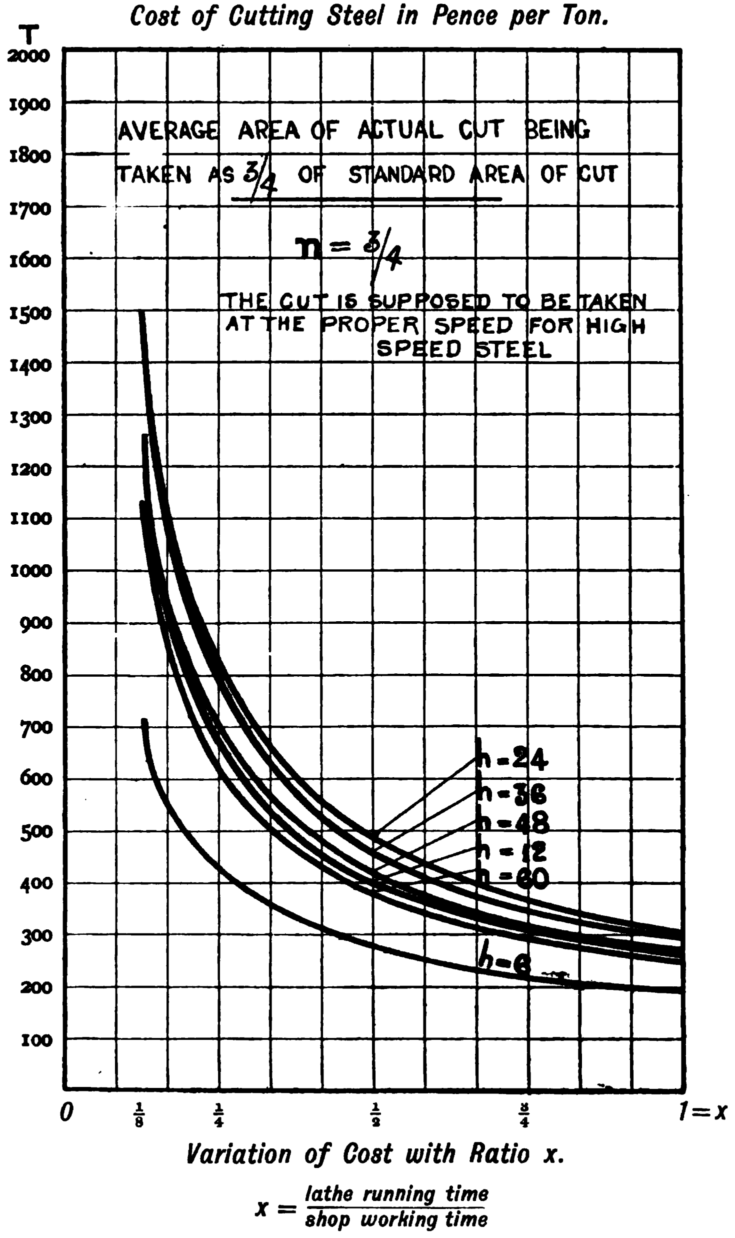
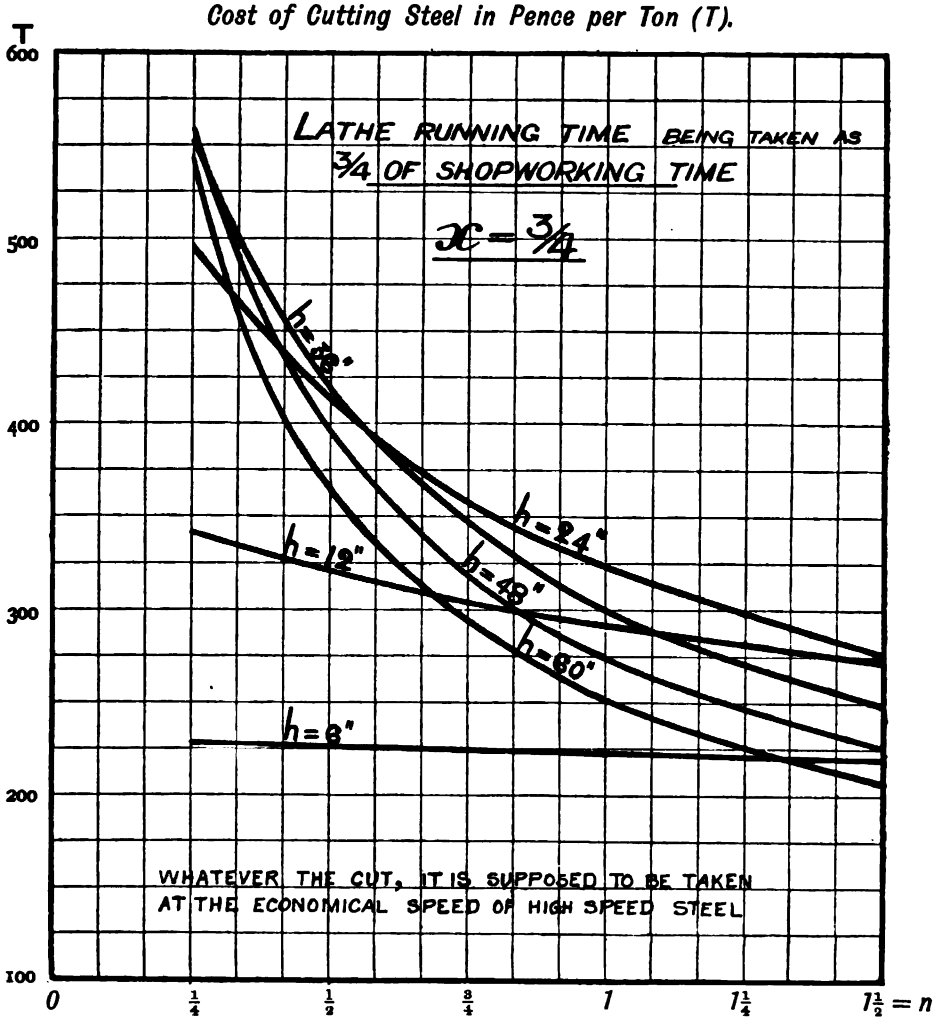


FIG. 237.

varying the number of hours per annum which the lathe runs out of a total possible of 3000; this number being the assumed number of shop-working hours per year. The figure diminishes at first very rapidly, but afterwards more slowly for all sizes of lathe, as the hours run by the lathe become a larger and larger proportion of the total. In drawing these curves (from Table LXI.) it has been assumed throughout that the average area of the cut actually taken is three-quarters of the standard area $\left(\frac{h^2}{6400}\right)$ which we have postulated.

Cost of Cutting Steel in Pence per Ton (T).



Variation of Cost with Ratio n for Various Sizes of Lathe.

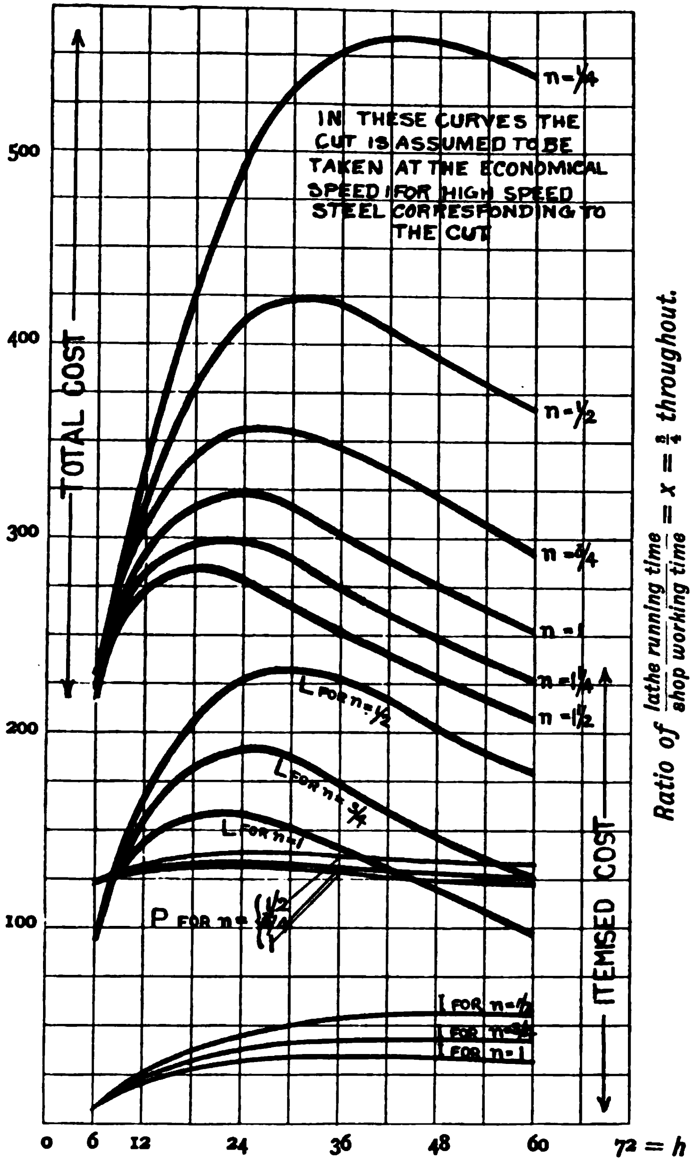
$$n = \frac{\text{average area of actual cut}}{\text{standard area of cut}}$$

FIG. 238.

In Fig. 238 we have taken it that the hours run by the lathe per annum are 2400, or three-quarters of the total possible; and have plotted (again from Table I.) the total cost in relation to the proportion which the average area of cut actually taken bears to the area we have assumed to be the standard. The cutting speed is assumed to be always the highest the steel can economically stand. We thus see that, as the average area of cut increases from 50 per cent. less than to 50 per cent. more than the standard, the cost per ton diminishes—

			d.		*d.*		*s.*	*d.*	
For a	6-in. lathe	from	226	to	217	or by	0	9	per ton.
„	12 „	„	321	to	272	„	4	1	„
„	24 „	„	412	to	276	„	11	4	„
„	36 „	„	418	to	249	„	14	1	„
„	48 „	„	395	to	226	„	14	1	„
„	60 „	„	367	to	208	„	13	3	„

Cost of Cutting Steel in Pence per Ton.



Variation of Total (and Itemised) Cost with Size of Lathe.

For various Values of $n = \frac{\text{average area of actual cut}}{\text{standard area of cut}}$

Fig. 239.

These figures show the savings to be effected by heavy as against light cuts all the year round. They are due to the greater weight-removing capacity of the high-speed steel when the heavier cuts are taken.

In Fig. 239 the total costs have been plotted (from Table LXI.) as functions of the size of lathe, for various ratios of average to standard areas of cut; the number of running hours being taken at 2400 out of 3000 throughout.

It is observed that, if the data upon which our estimates of cost are based be correct, the most uneconomical lathes are those of medium size, from 18-in. to 30-in. centres; the cost of cutting being 20 or 30 per cent. less when it is done upon quite small or very large lathes.

The costs have also been graphed in the same Fig. 239 in itemised form—first for labour, second for power, and third for interest and depreciation; the causes operating to produce the peculiar shape of total cost curve, shown above, being thus more clearly apparent.

The Cost of Cutting when the Area of Cut is not only less than the Standard Area, but is taken at some Speed less than the highest of which the Steel is economically capable.—This is what ordinarily happens in the case of the everyday use of a lathe. The extent to which it *must* happen with any given design of headstock is measured, as we shall show further on, by the ratio of the area of the "lathe-characteristic" to the area of the standard torque-speed diagram (*oabcd*, *vide* Diagrams I. to IX. below). This ratio we have called ζ; and in what follows we have found its value for a number of examples of lathes actually constructed. It now remains to show the effect upon the cost of cutting, which different values of this ratio ζ (*i.e.* different weight-removing capacities in a headstock) will produce.

As shown above, the greatest possible weight which can be removed when taking the standard cut is—

$$\begin{aligned} W_p &= 60 \times a_s\left(\frac{1}{a_s} + 15\right) \times 12 \times 29 \\ &= 207 + 3100a_s = 207 + \frac{3100h^2}{6400} \\ &= 200 + \frac{h^2}{2} \text{ (approximately)} \quad . \quad . \quad . \quad . \quad . \quad (173) \end{aligned}$$

lbs. per running hour.

The actually removed weight by a given headstock having a ratio ζ, will, by the definition of that ratio, be—

$$W_a = \zeta\left(200 + \frac{h^2}{2}\right) \text{ lbs. per running hour} \quad . \quad . \quad . \quad (174)$$

The weight removed per hour of shop-working time, when x is the fraction of actual hours run per year to total possible, is—

$$w_a = xW_a = x\zeta\left(200 + \frac{h^2}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad (175)$$

Hence the total cost of removing one ton of steel shavings by means of

TABLE LXI.

COST OF CUTTING STEEL IN PENCE PER TON.

As depending upon Size of Lathe, Hours run per Annum, and Average Area of Cut taken.

The cutting speed is supposed to be always the greatest the high-speed steel can economically take.

		n = 1½	n = 1¼	n = 1					n = ¾					n = ½					n = ¼				
		x = ¾	x = ¾	x =					x =					x =					x =				
				1	¾	½	¼	⅛	1	¾	½	¼	⅛	1	¾	½	¼	⅛	1	¾	½	¼	⅛
h = 6 in.	P	122	122	122	122	122	122	122	122	122	122	122	122	123	123	123	123	123	123	123	123	123	123
	L	87	89	68	90	135	271	542	69	92	136	270	540	71	94	142	284	568	72	96	145	289	578
	I	9	9	7	9	13	27	54	7	9	13	27	53	7	9	14	28	56	7	10	14	28	56
Total .	T	218	220	197	221	270	420	718	198	223	271	419	715	201	226	279	435	747	202	229	282	440	757
h = 12 in.	P	126	127	128	128	128	128	128	130	130	130	130	130	131	131	131	131	131	132	132	132	132	132
	L	128	136	109	145	218	436	872	116	155	233	465	929	125	167	250	500	1000	135	181	271	541	1082
	I	18	19	15	20	30	61	121	16	22	33	65	129	18	23	35	70	139	19	25	38	76	152
Total .	T	272	282	252	293	376	625	1121	262	307	396	660	1188	274	321	416	701	1270	286	338	441	749	1366
h = 24 in.	P	126	128	130	130	130	130	130	133	133	133	133	133	138	138	138	138	138	145	145	145	145	145
	L	125	140	121	162	242	484	968	142	190	284	568	1138	172	229	344	688	1376	217	289	434	868	1736
	I	25	28	24	33	48	96	192	28	38	56	112	224	34	45	68	136	272	43	58	86	172	344
Total .	T	276	296	275	325	420	710	1290	303	361	473	813	1495	344	412	550	962	1786	405	492	665	1185	2225
h = 36 in.	P	124	125	128	128	128	128	128	132	132	132	132	132	137	137	137	137	137	149	149	149	149	149
	L	101	117	105	139	208	418	835	129	172	258	515	1035	169	226	338	676	1351	245	326	490	980	1960
	I	25	28	25	34	50	101	202	31	42	62	125	250	41	55	81	163	326	59	78	118	236	472
Total .	T	250	270	258	301	386	647	1165	292	346	452	772	1417	347	418	556	976	1814	453	553	757	1365	2581
h = 48 in.	P	122	123	125	125	125	125	125	129	129	129	129	129	135	135	135	135	135	148	148	148	148	148
	L	82	96	88	117	175	350	700	111	148	222	445	890	153	203	305	611	1223	242	322	484	970	1923
	I	23	27	24	33	49	97	195	31	41	62	124	248	43	57	85	170	340	67	90	135	270	540
Total .	T	227	246	237	275	349	572	1020	271	318	413	698	1267	331	395	525	916	1698	457	560	767	1388	2611
h = 60 in.	P	119	121	123	123	123	123	123	126	126	126	126	126	132	132	132	132	132	146	146	146	146	146
	L	68	81	74	98	148	295	590	95	127	191	381	762	134	179	268	536	1072	227	302	453	907	1820
	I	21	25	23	31	46	92	184	30	40	60	119	238	42	56	84	168	335	71	94	142	284	568
Total .	T	208	227	220	252	317	510	897	251	293	377	626	1126	308	367	484	836	1539	444	542	741	1337	2534

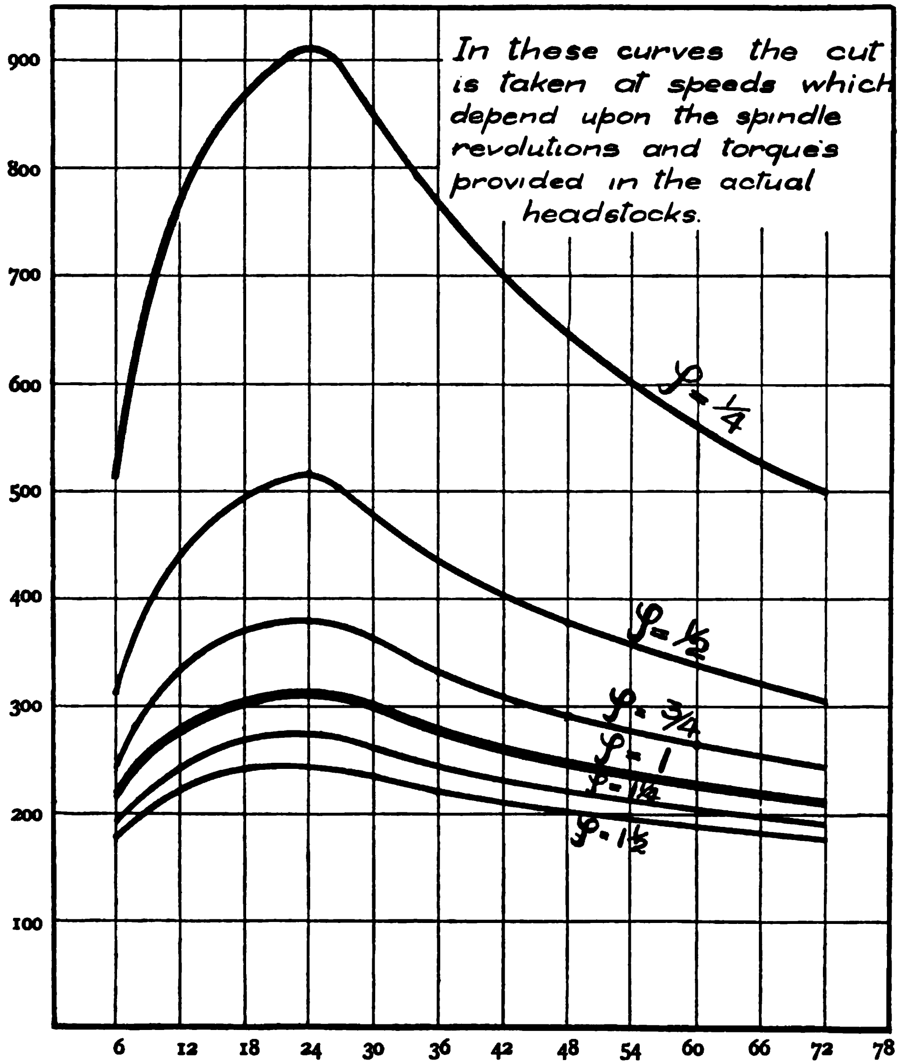
the given headstock, whose "characteristic" or weight-removing efficiency has a given value ζ, will be—

$$T = P_a + L_a + I_a$$

$$= \left[112 + \frac{336h}{W_a}\right] + \left[\frac{2240h}{w_a}\right] + \left[\frac{2240C}{125w_a}\right] \quad . \ . \ . \ . \ . \ . \ . \quad (176)$$

$$= 112 + \frac{336h}{\zeta\left(200 + \frac{h^2}{2}\right)} + \frac{2240h}{x\zeta\left(200 + \frac{h^2}{2}\right)} + \frac{2240C}{125x\zeta\left(200 + \frac{h^2}{2}\right)} \quad (177)$$

Cost of Machining Steel in Pence per Ton for various Sizes of Lathe.



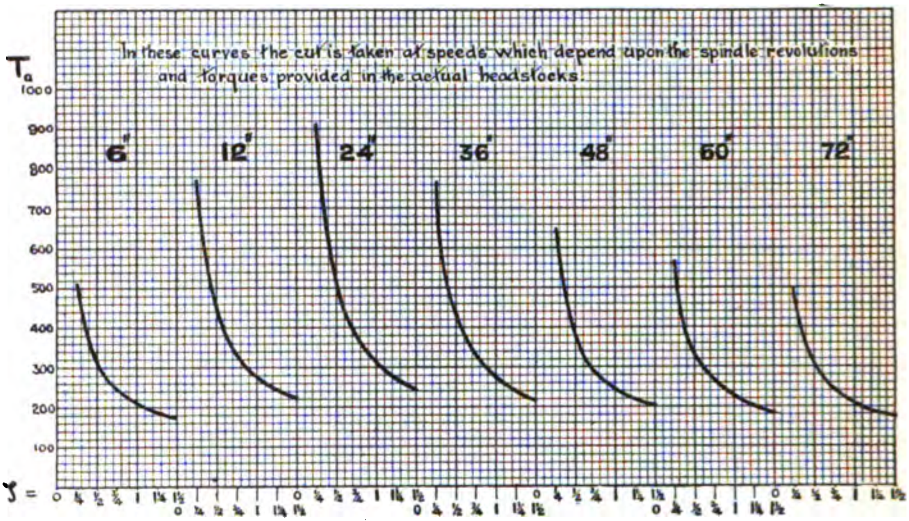
In these curves the cut is taken at speeds which depend upon the spindle revolutions and torques provided in the actual headstocks.

Variation with Different Values of the Ratio ζ = Weight Removable $\left\{\begin{matrix}\text{by a Given}\\ \text{by a Perfect}\end{matrix}\right\}$ Headstock.

FIG. 240.

We have calculated out the values of T for various sizes of lathe, from 6-in. to 72-in. height of centres; and for six different values of ζ, viz.:— $\zeta = \frac{1}{4}, \frac{1}{2}, \frac{3}{4}, 1, 1\frac{1}{4}$, and $1\frac{1}{2}$. The lathe running time has been taken equal to 2400 out of a possible 3000 hours per annum (*i.e.* $x = \frac{4}{5}$); the same for all sizes of lathe (possibly a smaller value ought to have been taken for x in the case of small lathes than for large ones).

Cost of Machining Steel in Pence per Ton T_a.



Variation (for each size of Lathe) with different Values of ζ = Weight removable $\left\{\frac{\text{by a given}}{\text{by a perfect}}\right\}$ Headstock.

FIG. 241.

The results are given in Table LXII., and are plotted in Figs. 240 and 241.

TABLE LXII.

Cost of Machining Steel in Pence per Ton.

As depending upon Headstock Efficiency.

x = value of $\frac{\text{running}}{\text{working}}$ hours per annum, taken = ¾ throughout.

The cutting speeds here depend on the spindle revolutions and torques actually provided in the headstock.

		ζ =					
		1½	1¼	1	¾	½	¼
h = 6 in.	P	118	119	121	124	131	149
	L	55	66	82	109	164	328
	I	6	7	9	11	17	34
Total . .	T	179	192	212	244	312	511
h = 12 in.	P	122	124	127	131	142	171
	L	88	105	132	176	263	526
	I	13	15	19	26	38	77
Total . .	T	223	244	278	333	443	774
h = 24 in.	P	123	125	129	134	145	178
	L	103	123	154	206	307	618
	I	20	24	30	40	60	120
Total . .	T	246	272	313	380	512	916
h = 36 in.	P	122	123	126	131	140	169
	L	76	94	117	155	233	468
	I	21	25	32	42	63	126
Total . .	T	219	242	275	328	436	763
h = 48 in.	P	120	122	124	128	136	160
	L	61	74	91	122	182	368
	I	20	24	30	41	60	120
Total . .	T	201	220	245	291	378	648
h = 60 in.	P	119	120	122	125	132	152
	L	49	59	69	99	148	296
	I	19	23	29	39	58	116
Total . .	T	187	202	220	263	338	564
h = 72 in.	P	118	119	121	124	129	147
	L	40	49	62	82	123	247
	I	18	22	27	36	54	109
Total . .	T	176	190	210	242	306	503

CHAPTER XL

THE TORQUE-SPEED DIAGRAM. LATHE CHARACTERISTICS

It has long been a desideratum on the part of the lathe-user to have a method of comparison by which the values of the different types of headstock, and of the various factors entering into their design as power transmitters, could be fairly estimated.

In reference to this matter, Mr. P. V. Vernon, of Messrs. Herbert, makes the following remarks:—[1]

"Several suggestions have been made recently as to establishing a standard for estimating the power of headstocks, and most of the suggestions have been based upon the maximum pulley diameter and the maximum gear ratio. Consideration of the points discussed above as to the relations between speed and power, however, makes it clear that such a standard would be misleading, as in the majority of cases, especially with high-speed steel, the maximum gear ratio is not in use, and in very many cases the belt is not upon the largest step of the cone, when the required work is to be done.

"In stating the strength of a chain, consideration must be paid to the strength of the weakest link, and the same course should be adopted when estimating the value of a machine tool for any given work, and the idea, therefore, almost suggests itself that it would be as reasonable to estimate the power of a lathe by the minimum values of cone pulley diameter and gear ratio as by their maximum values. In estimating the power of a lathe for any work it is necessary to take into consideration that part of the speed range at which the work must be done in order to get the required cutting speed suitable for the tool steel used, and on this account it seems that a better basis can be arrived at from the quantity of belt passing over the pulley (width multiplied by velocity) per inch or foot of cut at certain specified cutting speeds on the various diameters. Such a method takes into account the various efficiencies on the different steps of the cone pulley, and also does not encourage the formation of a mistaken estimate based on a high ratio of gearing which could never be used with the high-speed steel tools."

The question is, What are the economic values to the purchaser of the various items in a lathe specification? Such as—*The drive*, whether by belt or motor, by cone on or off the spindle, or by an all-gear head. The spindle speeds and their ratio. The number and ratio of the back gears. The number and arrangement of the counter shaft speeds. The ratios of the successive spindle speeds, as governed by the steel to be employed, and the size of cut and of job.

---

[1] *The Engineer*, Mach. Tool Supplement, Nov. 4, 1904, p. iii.

The problem is much more complex than is generally supposed; and no single simple rule, such as that above referred to by Mr. Vernon, will be found sufficient.

A method which we have devised, and propose now to explain, appears to us likely to prove useful for the purpose in view. By its employment, together with the cost curves of Chapter XXXIX., it would seem possible to estimate the value of any given design of headstock relatively to a standard of reference, and to express this value numerically by its effect upon the cost of machining unit weight of the metal to be operated upon.

The method depends essentially upon the plotting of a torque- (spindle) speed diagram of constant value for the given headstock, and the comparison of this with a similarly constructed curve for a lathe without friction (or other) losses, and with no gaps in the succession of possible spindle speeds.

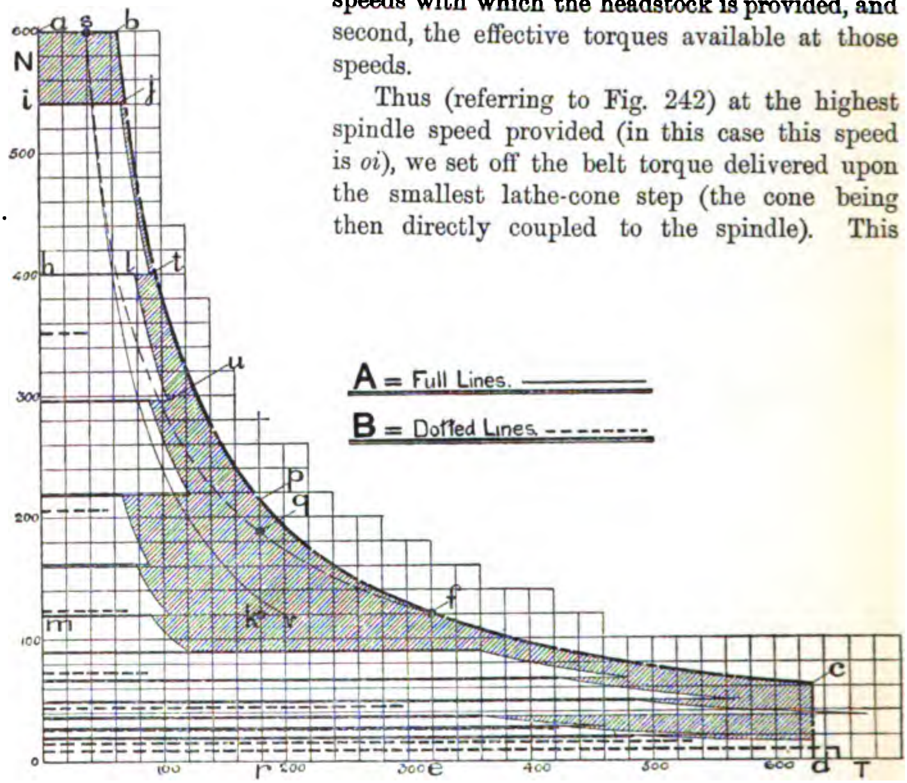
The spindle revolutions are set off as ordinates on a base of the torques of which the lathe is capable. Thus the standard cut $a_s = \frac{h^2}{6400}$, when taken at face-plate radius h, requires a torque $T_s = \frac{fa_sh}{12}$ ft. lbs. ($f = 100 \times 2240$ lbs.), and this is set off to some convenient scale, as *od* lathe "A," Fig. 242. This cut can be taken at a speed $v_s = \frac{1}{a_s} + 15$ with high-speed steel without undue waste of tool steel. The spindle revolutions N_s, to give this speed on a radius h'', are got from $\frac{2\pi h}{12}N_s = v_s$, or $N_s = \frac{6}{\pi h}v_s = \frac{6}{\pi h}\left(\frac{1}{a_s} + 15\right)$. Hence, on some convenient scale set off $dc = N_s$.

If the standard cut a_s be required on some smaller radius r, but at its proper speed $v_s = \left(\frac{1}{a_s} + 15\right)$, then the corresponding torque T is $\frac{far}{12}$ ft. lbs. Thus $\frac{T}{T_s} = \frac{r}{h}$. The new spindle revolutions N corresponding to the smaller radius are $N = \frac{6}{\pi r}\left(\frac{1}{a_s} + 15\right)$, and $\frac{N}{N_s} = \frac{h}{r}$. Hence $\frac{TN}{T_sN_s} = \frac{r}{h} \cdot \frac{h}{r} = 1$, or the product of the torque and spindle revolutions when the standard area of cut is taken at its proper cutting speed is the same for every job-radius. If, therefore, we construct a rectangular hyperbola *cb* on the axes of torque and revolutions, any point, such as *f*, upon it will give $ef = N$, the correct spindle revolutions, and $oe = T$, the proper torque, when the standard cut a_s is taken on a job of radius r, such that $\frac{oe}{od} = \frac{r}{h}$. If, now, some smaller cut $na_s (n < 1)$ is to be taken on a job of radius r, the required torque T' will be less in proportion, and $\frac{T'}{T} = \frac{na_s}{a_s} = n\left[\frac{mk}{mf}, \text{say}\right]$. The tool steel can, however, take this lighter cut at a higher speed than that obtained with N-spindle revolutions on this job of radius r. If the speed of cutting could increase as fast as the area diminished, the new spindle revolutions *rp* would be given by the point *p* (where the vertical through *k* cuts the hyperbola *bc*).

The tool steel would soon be ruined, however, by this excessive speed, and the truly economical cutting speed, as obtained by experiment and indicated on Fig. 2, p. 9 (of the Manchester committee's trials), should be substituted. In this way the thin dotted curve *fq* is obtained, and the proper spindle revolutions are given by the height *rq*. By means of this curve, the hyperbola *cb* and the curves *fq*, drawn for those job diameters which are most likely to occur in the life of the given lathe, we can represent those limitations to the capacity of the lathe which are imposed by the total power it receives and by the brand of tool-steel it is intended to employ.

In order, now, to compare the lathe as actually constructed with the perfectly efficient and fully capacious hypothetical tool just described, we must set off, on the same diagram, first, the values of the various spindle speeds with which the headstock is provided, and second, the effective torques available at those speeds.

Thus (referring to Fig. 242) at the highest spindle speed provided (in this case this speed is *oi*), we set off the belt torque delivered upon the smallest lathe-cone step (the cone being then directly coupled to the spindle). This is found by multiplying the belt width by the effective tension per inch of width and by the smallest cone-step radius. The quantity plotted as *ij* is thus obtained, and it is seen to be, in this case, not much below the torque required by the standard curve *bc*. This shows that if this belt torque could be conveyed without frictional losses through the gears back to the spindle, when cutting with the back gear at work, the full capacity of the steel on a large job at a slow speed could be almost realized.



Torque Speed Diagram for 6-in. Lathe (cone on spindle).

FIG. 242.

The friction losses, however, cause a considerable diminution of the effective or tool torques at the lower spindle speeds, and they are accordingly seen to fall proportionally much further short of curve *bc* than *ij* does.

When the belt is shifted on to the next larger lathe-cone step, the torque is increased, because the belt-pull is the same (it really ought to be somewhat diminished because its speed drops) and the step radius is larger. The value obtained is plotted, as *hl*, opposite to its corresponding spindle speed, 400 revolutions per minute. The product of the torque and speed is seen, however, to be considerably less for this case than for the former. This is because the spindle speed diminishes at a greater rate than the torque increases.

Proceeding in this way, we may construct the "characteristic" diagram of the given headstock for the whole range of speeds of which the spindle is capable. We connect these torques and speeds by bits of hyperbolas (as *jt*, *lu*, etc.) for reasons to be explained below, and the diagonally hatched area so obtained is found to be a measure of the economic value of the given design.

Torque and Revolution Chart.—This chart (*vide* folding-plate in cover) has been drawn to facilitate the determination of the values of the forces and torques required for taking any given area of cut upon machinery steel of various diameters by means of a lathe, and for finding the number of revolutions per minute at which the spindle of the lathe must run in order that the cutting speed may be obtained, when taking the cut on those diameters, which is considered suitable to the given area of cut and the kind of tool steel employed.

In particular, we have shown upon the chart (by the two thick black curves), the relations between size of lathe and standard area of cut, and between size of lathe and spindle speed for standard cutting speed which we postulated in Chapter VII., pp. 32 and 33, *ante*.

These rules are—

$$a_s = \frac{h^2}{6400} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

and

$$v_s = \frac{1}{a_s} + 15 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The cutting force necessary to be applied to remove a cut of area a_s on a mild or medium steel job is—

$$F_s = fa_s = f\frac{h^2}{6400} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (178)$$

where f is the cutting pressure, say in lbs. per sq. in.

On p. 33, *ante*, we took for f the value 115 tons per sq. in.; but as we now propose to compare the cutting force required at the tool with the *net* force which can be delivered there by the lathe when allowances have been made for frictional losses in the headstock, it will be more correct to take a lower value than the above. A good average value for steel is 100 tons per sq. in. (*vide* Fig. 8, p. 17), and then

$$f = 224000 \text{ lbs. per sq. in.}$$

and

$$F_s = 224000\frac{h^2}{6400} = 35h^2 \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (179)$$

On chart ($F - r$) this quantity is plotted as the curve GH. Its ordinates (measured downwards) are the values of F_s, and its abscissæ (measured leftwards) are the corresponding values of h.

Thus for h (or r), equal to 10 inches, $F_s = 3500$ lbs.; and this value is found on the diagram by following the vertical KL downwards till it cuts GH in L. A horizontal, LM, through L, gives 3500 lbs. on the F-scale.

The product of KL and LM gives Fr, the net torque, here 35000, which the lathe is overcoming in inch-lbs., when the standard cut is being taken on face-plate diameter.

The same torque will be developed by the lathe if the area of the cut (and also, of course, F) is increased proportionally as the radius of the job is diminished. Thus at 5-in. radius the cutting force may be 7000 lbs. (as shown by points *klm*). A rectangular hyperbola connecting all such points as L and l will therefore give the various cutting forces (and therefore areas of cut) which the given lathe should be capable of, on the corresponding job diameters, without the standard torque being exceeded (here $\frac{35000}{12} = 2916$ ft. lbs.), represented by the rectangle KLMO.

Further, the numerical value of this torque can be read off on the F-scale *in ft. lbs.* by noticing the point where the hyperbola through L cuts the strong line TT passing through $r = 12$ inches.

Alongside of the F-scale there are indicated for convenience the areas of cut which will require the cutting forces marked opposite to them to be applied when operating on mild or medium steel.

As indicated above, the $(F - r)$ chart is also intended for finding the torques when any other rule for fixing the standard cut is followed.

Thus, let it be required to take a cut $\frac{3}{8}$ by $\frac{1}{16}$ on a medium cast-iron job of 10 inches diameter. Then following the ordinate $r = 5$ inches downwards until it passes through the abscissa $a = 0{\cdot}0235$ sq. in., we find that the intersection lies halfway between the constant torque hyperbolas marked 2000 and 2500 ft. lbs.

The actual torque is $\frac{Fr}{12} = T = 2190$ ft. lbs. This value may be read off at once by following along between the two hyperbolas mentioned until they cross the vertical line TT at $r = 12$ inches. This gives the torque when operating upon machinery steel. To find the value for cast iron, we turn to Fig. 9, p. 17, where the net cutting force is plotted for various areas of cut and angles of tool.

For $a = 0{\cdot}0235$ and a tool angle of 80°, we find $f = 58$ tons. The torque obtained above must therefore be multiplied by $0{\cdot}58$ to get the correct value in this case. We thus find $T = 2190 \times 0{\cdot}58 = 1270$ ft. lbs.

To determine the spindle revolutions at which the required cut should be taken on medium cast iron, all we have to do is to refer to Fig. 3 on p. 9, where we find that a cut of 0·0235 sq. ins. area can be removed at 38 feet per minute (the tool then lasting 20 minutes). If we say 35 feet per minute to be safe, then to find the spindle revolutions from chart $(N - r)$, we follow between the hyperbolas of constant speed 30 and 40 to where they cross the vertical through $r = 5$ inches. The corresponding spindle speed is 17 revolutions per minute.

The $(N - r)$ chart thus provides facilities for determining the proper spindle speed which the lathe must run at in order to give any desired cutting speed upon work of a given diameter.

In particular, the heavy curve AB shows the relation between spindle revolutions and height of lathe centres which must obtain when the standard cut is taken at its proper speed for steel. Referring first to this, taking the rules Nos. (27) and (28) given above, and combining them, we obtain—

$$v_s = 15 + \frac{6400}{h^2} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (180)$$

This gives the "standard cutting speed" v_s (feet per minute) for a lathe of h inches height of centres. The lathe must be capable of giving this speed (which is that for which a high-speed tool will last for a reasonable time when taking the standard cut corresponding to the given size of lathe) when working on some diameter of job to be agreed upon. On p. 35 we have postulated this diameter to be that of the face-plate ($= 2h$ in.); but this may not meet with universal acceptance, even for a pure high-speed lathe. In cases where it is agreed to, the number of spindle revolutions per minute (N_s) which must be made in order to produce a speed v_s on a diameter $2h$ is obviously—

$$N_s = \frac{12v_s}{\pi(2h)} = \frac{12}{\pi(2h)}\left[15 + \frac{6400}{h^2}\right]$$

or

$$N_s = \frac{12{,}800}{h^3} + \frac{30}{h} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (32) \text{ (comp. p. 35)}$$

The values of this expression have been plotted as ordinates on a base of height of centres h (or r) as the curve AB on chart ($N - r$).

Thus on a 12-in. centres lathe (for which the standard cut is 0·0225 sq. in. area, and the corresponding cutting speed 59·5 feet per minute) the spindle speed, N_s, which will give the standard linear cutting speed v ($= 59{\cdot}5$) is 9·5 revolutions per minute. This agrees with the co-ordinates of the point D, which is the intersection on the curve AB of the vertical through $h = 12$ inches. The same cutting speed can, of course, be obtained on any other radius if the spindle speed be made to vary inversely as the radius. Thus, if $r = 4$ inches, $N = 28{\cdot}5$ as got from point d, on the vertical cd through $r = 4$ inches, where $\frac{cd}{\mathrm{CD}} = \frac{\mathrm{ED}}{ed}$. The points D and d are therefore situated on a hyperbola of constant cutting speed ($v_s = 59{\cdot}5$). A number of similar hyperbolas have been drawn on the chart, and the values of the cutting speeds they represent marked upon them.

This ($N - r$) chart may therefore be used for finding the spindle revolutions required with any given job radius to produce any other cutting speed than that given by rule (32). Thus, let it be required to find the spindle speed which will give a cutting speed of 40 feet per minute on a job of 10 inches diameter. Then, following the vertical through $r = 5$ inches, up to where it intersects the hyperbola of 40 feet cutting speed we find $N = 15\frac{1}{4}$.

It may be observed that, since $2\pi r\frac{N}{12} = v \therefore N = v$ if $r = \frac{6}{\pi} = 1{\cdot}912$ inches. Thus by noting where the strong vertical line VV through $r = 1{\cdot}912$ inches cuts the various hyperbolas, the constant cutting speeds they represent can be read off directly on the N-scale.

We are now in a position to draw a torque-revolution ($T - N$) diagram for a lathe of any given height of centres. The product of the greatest torque of which a lathe is capable into the spindle revolutions at which it can be taken is clearly a measure of the power of the lathe.

$$[\text{For } 33{,}000 \text{ CHP} = Fv = \frac{F2\pi Nr}{12} = \frac{\pi}{6}(Fr)N = \frac{\pi}{6}TN]$$

In fact, TN = 63,024 CHP, where T is the torque in foot-lbs., N the corresponding spindle revolutions, and CHP the net (or cutting) horsepower.

We might, therefore, again describe a number of rectangular hyperbolas having N for ordinates and T for abscissæ, and mark each of them with its appropriate horse-power as depending on the sizes and desired capacities of lathes.

It would appear to be sufficient, however, for the purpose of studying any given case to set off the torque and spindle revolutions from the $(F - r)$ and $(N - r)$ charts already given, as and when required.

An example will show how simply we may proceed.

To draw the torque-speed diagram for a 6-in. lathe, assuming the standard area and cutting speed to be given by our rules.

On the $(F - r)$ chart, the intersection of the vertical through $r = 6$ inches with the GH curve gives 1260 lbs. (area of cut = 0·0056 sq. in.) as the standard cutting force F_s. The intersection of the hyperbola through this point (6, 1260) with the 12-in. vertical (TT) gives 630 ft.-lbs. as the standard torque.

On the $(N - r)$ chart the intersection of the vertical through $r = 6$ inches with the curve AB gives $N_s = 61{\cdot}2$ revolutions per minute, as the spindle-speed at which the standard cutting speed (corresponding to area 0·0056 sq. in.) will be obtained on a diameter of 12 inches, equal to that of the face-plate.

We therefore set off on Fig. 242 the point *c*, having 61·2 for ordinate and 630 for abscissa; and through it draw a rectangular hyperbola, having the axes of T and N for assymptotes. The greatest value of N is $\frac{3600}{h}$ or $Ng = 600$; the lowest value of N is 61·2 (*vide* Table XXVI., p. 36) for a high-speed steel lathe.

The hyperbola need not therefore be extended beyond the points $oa = 600$ and $od = 630$.

Any point (*f* say) on this hyperbola will give, by its co-ordinates, *oe* and *ef*, the torque (in foot-pounds) of which the lathe should be capable, and the corresponding spindle-speed, *if the available net* (or useful) *horse-power were the same throughout its range, and if every possible spindle-speed between* Ng *and* N_l *were available.*

Given, now, the size of belt, the diameters of cone steps, and the gear arrangements of the headstock of the lathe we are examining, we must inquire whether, and to what extent, its belt can deliver through the gears the torques indicated by the above hyperbola *bc* at all the various spindle speeds.

If we neglect friction the power given out by the belt equals that absorbed by the tool. In that case we have—belt-torque × cone revolutions = tool-torque × spindle revolutions. That is—

$$PB\frac{E}{2 \times 12}n = \frac{Fr}{12}N = TN \text{ if } r = 12\text{-in.}$$

$$\therefore T = PB\frac{E}{24}\ \frac{n}{N}\text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (181)$$

Here T = torque at tool in foot-lbs.
P = effective belt pull per inch of width.
B = belt width in inches.
(A, B, C . . .) E = cone step diameters in inches.
n = cone revolutions per minute.
N = spindle revolutions per minute.

When the cone is directly coupled to the spindle n = N, and we have—

$$T = PB\frac{E}{24}\text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (182)$$

When a back-gear of ratio R is interposed between cone and spindle, we have $n = RN$; in which case—

$$T = PB\frac{E}{24}\ \frac{1}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (183)$$

If the countershaft speed (or speeds) and the proportions of the cone step are given, then the various spindle speeds are known. Thus, for one countershaft speed, four cone steps and one back-gear—

$$\begin{array}{lcl}
N_1 = \dfrac{cA}{D} & \text{and as above} & T_1 = \dfrac{(PB)D}{24} \\
N_2 = \dfrac{cB}{C} & ,, & T_2 = \dfrac{(PB)C}{24} \\
N_3 = \dfrac{cC}{B} & ,, & T_3 = \dfrac{(PB)B}{24} \\
N_4 = \dfrac{cD}{A} & ,, & T_4 = \dfrac{(PB)A}{24} \\
 & & \\
N_5 = \dfrac{cA}{DR} & ,, & T_5 = \dfrac{(PB)D}{24R} \\
N_6 = \dfrac{cB}{CR} & ,, & T_6 = \dfrac{(PB)C}{24R} \\
N_7 = \dfrac{cC}{BR} & ,, & T_7 = \dfrac{(PB)B}{24R} \\
N_8 = \dfrac{cD}{AR} & ,, & T_8 = \dfrac{(PB)A}{24R}
\end{array}$$

Here c = the countershaft revolutions (here one speed only); A, B, C, and D are the cone step diameters (here four in number); and R is the back-gear ratio. (PB) = effective belt pull lbs.

In a particular example (named B, Fig. 242, also Diagram I., by us) of a lathe of 6-in. centres made by a well-known firm, the four cone steps were of 5-in., 7-in., 9-in., and 11-in. diameter, each 2¾ in. wide. The countershaft speed (c) was 160 revolutions per minute, and the back-gear ratio (R) was 8 to 1.

We have, therefore, for this lathe—

$$
\begin{array}{llll}
N_1 = \dfrac{160 \times 11}{5} = 352 & \text{and} & T_1 = \dfrac{5 \times 75 \times 2\frac{1}{2}}{24} & = 39 \\[2ex]
N_2 = \dfrac{160 \times 9}{7} = 206 & \text{,,} & T_2 = \dfrac{7 \times 75 \times 2\frac{1}{2}}{24} & = 55 \\[2ex]
N_3 = \dfrac{160 \times 7}{9} = 124 & \text{,,} & T_3 = \dfrac{9 \times 75 \times 2\frac{1}{2}}{24} & = 70 \\[2ex]
N_4 = \dfrac{160 \times 5}{11} = 73 & \text{,,} & T_4 = \dfrac{11 \times 75 \times 2\frac{1}{2}}{24} & = 86
\end{array}
$$

$$
\begin{array}{llll}
N_5 = \dfrac{160 \times 11}{5 \times 8} = 44 & \text{,,} & T_5 = \dfrac{0{\cdot}9 \times 8 \times 2\frac{1}{2}}{24} & = 296 \\[2ex]
N_6 = \dfrac{160 \times 9}{7 \times 8} = 26 & \text{,,} & T_6 = \quad \text{,,} & = 418 \\[2ex]
N_7 = \dfrac{160 \times 7}{9 \times 8} = 15{\cdot}5 & \text{,,} & T_7 = \quad \text{,,} & = 531 \\[2ex]
N_8 = \dfrac{160 \times 5}{11 \times 8} = 9 & \text{,,} & T_8 = \quad \text{,,} & = 650
\end{array}
$$

It will be noticed that for T_5 to T_8 we have multiplied by 0·9. This is to allow for the frictional loss due to the two pairs of gears interposed between cone and spindle when the back-gear is in operation, the efficiency of each pair being taken at 0·95. Strictly speaking, some allowance should be made if only for journal friction upon the first four speeds also, but we have neglected this.

These results have been plotted on Diagram I., and are shown as horizontal dotted lines.

Inspection shows at once how very much the lathe, as constructed, falls short of the standard requirements we have postulated; notwithstanding the fact that we have allowed an effective tension of 75 lbs. per inch of width, which is nearly double of the proper value, for such small pulleys.

CHAPTER XLI

THE TORQUE-SPEED CHARACTERISTIC OF A LATHE—ITS PROPERTIES—ITS VALUE AS AN INDEX OF THE CAPABILITIES OF THE HEADSTOCK FOR ECONOMICAL CUTTING

In order to obtain in a concrete form, and not merely qualitatively, the economic value of different designs of lathe, as affected by the number and ratio of the different spindle speeds and the net torques available for actual cutting at each, we propose to consider the torque-speed diagram somewhat more closely, and to discover what relations exist between the form and area of that diagram and the weight which can be machined off per unit of time by the lathe to which it belongs.

As we have already explained, the hyperbola *bc* (Figs. 242 and 243) shows the relation which must exist between torque and spindle speed if the lathe is to be able to take the standard area of cut on every diameter of job within the compass of the machine. Thus *o'c* (Fig. 243) represents the torque needed to take that cut at radius h; therefore *mf* (half of *od*), which measures the torque at spindle revolutions *ef* (twice *dc*), corresponds to the standard cut taken at radius $r = \frac{1}{2}h$, whilst *o'f'* means a cut of area half the standard taken at radius $= h$. In what follows, we shall denote these two different cuts, both requiring the same torque by the equal but differently situated abscissæ *mf* and *o'f'*. Thus *mf* will represent the standard cut a_s taken at radius $r = \frac{mf}{od}h$, whilst *o'f'* will stand for a cut of area $a = \frac{o'f'}{oc} \times a_s$ taken on radius $h = \frac{o'c}{od}h$. In either case the power required is the same, and is equal to that represented by *oe* . *ef* or *od* . *dc*. The rectangles *oe* . *ef* and *od* . *dc* are equal, and they represent not only the product of torque and speed (*i.e.* the power), but also the weight removed per unit of time.[1] In the same way, if any other cut whose area is to that of a cut requiring torque *mf* as *mk* is to *mf* be taken at spindle speed *ef*, then the weight cut per unit of time will be proportional to rectangle *or* . *rk*. Now, if the lathe were capable of every possible speed between *oa* and *dc*, and if the tool steel used could stand it, the given cut *mk*

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[1] For the weight cut per running hour is $W = 60 \times 12\mu av = 207av$.

Now

$$TN = \frac{FrN}{12} = \frac{farN}{12} = \frac{far \times 6v}{12\pi r} = \frac{f}{2\pi}av$$

$$\therefore W = 207 \times \frac{2\pi}{f}TN = \frac{207 \times 2\pi}{224{,}000}TN = \frac{TN}{1724} \text{ lbs. per hour.}$$

would be taken at spindle speed *rp* (*p* being the point on the hyperbola *bc* vertically above *k*), and the weight removed would be as *or . rp*, and would be thereby again raised to the standard rate of removal represented by *od . dc* or *oe . ef*.

If the cutting power of the steel were equally great at all speeds, that is, if the cutting speed could be raised at the same rate as the area of the cut is lowered, only one spindle speed would be necessary! Metal would then be removed at the same rate per hour, whether by, say, four light cuts at 40 feet per minute, or one heavy cut (four times as great) at 10 feet per minute. The only difference would be due to the loss of time in drawing back the saddle.

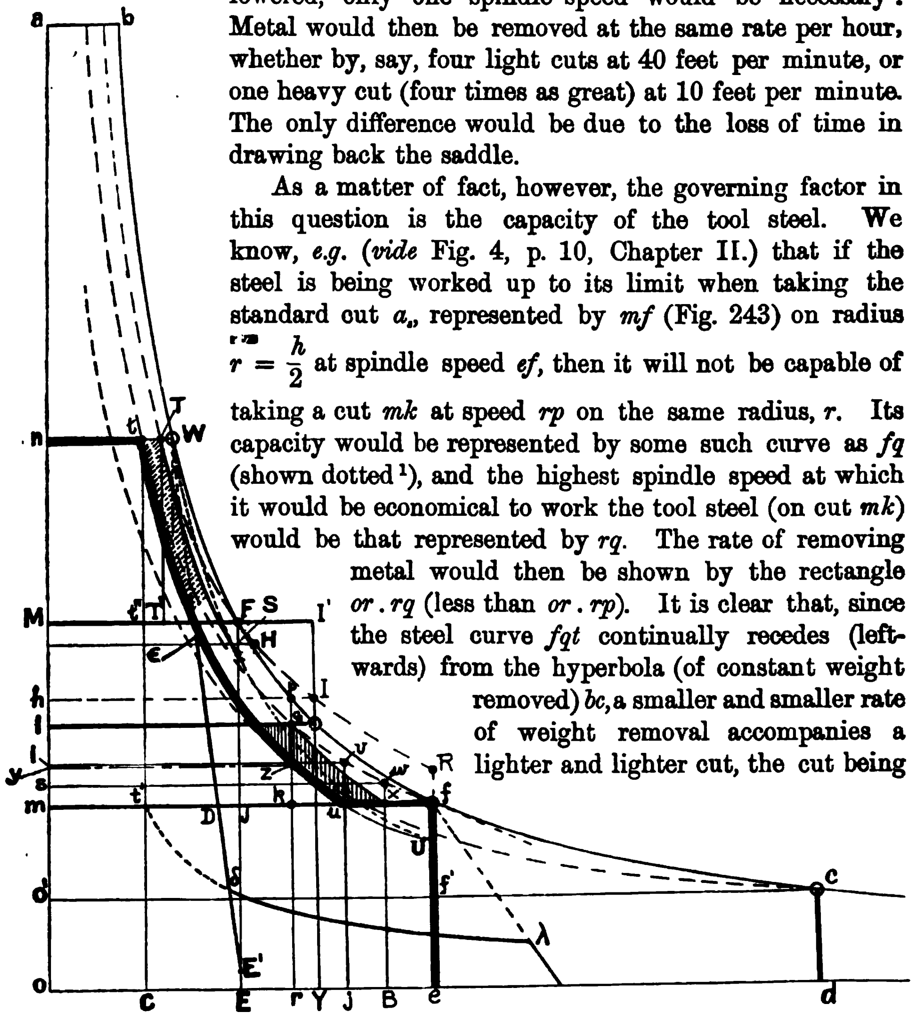

FIG. 243.

As a matter of fact, however, the governing factor in this question is the capacity of the tool steel. We know, *e.g.* (*vide* Fig. 4, p. 10, Chapter II.) that if the steel is being worked up to its limit when taking the standard cut a_s, represented by *mf* (Fig. 243) on radius $r = \frac{h}{2}$ at spindle speed *ef*, then it will not be capable of taking a cut *mk* at speed *rp* on the same radius, *r*. Its capacity would be represented by some such curve as *fq* (shown dotted[1]), and the highest spindle speed at which it would be economical to work the tool steel (on cut *mk*) would be that represented by *rq*. The rate of removing metal would then be shown by the rectangle *or . rq* (less than *or . rp*). It is clear that, since the steel curve *fqt* continually recedes (leftwards) from the hyperbola (of constant weight removed) *bc*, a smaller and smaller rate of weight removal accompanies a lighter and lighter cut, the cut being assumed to be always taken at the highest economic steel capacity. Thus a cut represented by *mt′* $\left[\text{of area } \frac{mt'}{mf}a_s\right]$ taken at spindle speed *on*, corresponds to a weight removed per unit of time proportional to the area of the rectangle *on*.*nt*. If through *t* and *q* we draw hyperbolas *tu* and *qx*, the axes being *oa* and *od*, the above area *on*.*nt* is the same as *om*.*mu*, which is less than *om*.*mx* (= *or*.*rq*), and much less than *om*.*mf*, the standard rate of weight removal. Thus the heavier the cut, the more metal can be removed per unit of time by a lathe having every possible spindle speed, because the weight-removing capacity of the steel is greater the heavier the cut.

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[1] To draw such a curve as *fq*, we proceed as follows: Ascertain from Figs. 2 (or 3), p. 9, the proper cutting speed v_s which high-speed steel can take, when removing the given "standard" area of cut a_s on steel (or cast iron) which corresponds to the size of lathe under discussion. Then this *cutting speed* v_s must be represented by the ordinate *ef* (Diag. 5).

Similarly *rq* must represent, on the same scale as *ef* measures v_s, the cutting speed v (obtained from Fig. 2) of which the tool steel is capable, when taking a cut of area $\frac{mk}{mf}a_s$.

This is obvious, for as both cuts are removed at the same radius, their areas are proportional to their torques, and if *mf* be assumed to measure a cut of standard area a_s, *mk* will measure the area of a cut $\frac{mk}{mf}a_s$.

We have, so far, only considered those cases in which the limiting weight which could be cut per unit of time was fixed by the tool steel, not by the lathe; for we assumed, in the discussion, that every possible spindle speed between *oa* and *dc* was available, and that torques represented by the abscissæ of the curve *bc* could always be counted upon at the tool. We must now take the actual case, and see what effects are produced upon the weight which can be removed, (*a*) by gaps in the spindle speeds, and (*b*) by torques less than the above required values.

(*a*) **Effect of Gaps in Speed Curve.**—Suppose, therefore, that the lathe has no spindle speed between those represented by *on* and *om* (Fig. 243), and that the same area of cut (*a*) represented by *mk* $\left(a = \frac{mk}{mf}a_s\right)$ has to be removed from a job of the same given radius as before $\left(r = \frac{oe}{od}h\right)$. Then, so far as the capacity of the steel is concerned, the cut could be taken at the highest speed *on* by first (twice) removing a depth proportional to *mt′* instead of *mk*, and then taking the rest of the required depth of cut off the bar at such an increased width of traverse that the area of cut taken the last time over is the same as that (represented by *mt′*) taken the first time over. The time rate of weight removal would then have been the same for all the cuts, and would be represented by *on*.*nt*.

On the other hand, the whole depth of cut might have been taken at once at the lower spindle speed *om*, and the rate of weight removal would then have been proportional to *om*.*mk*.

Through *t* draw the (heavy line) hyperbola *tu* (axes *oa* and *od*). Then, since *on*.*nt* = *om*.*mu*, and *om*.*mu* is greater than *om*.*mk*, we see that it is better to go more than once over and take the cut at the higher spindle speed *on* than to take it (in once) at the lower speed *om*.

This is true for all areas of cut corresponding to points on the horizontal line *mf* lying between *t′* and *u*.

Any cut greater than that represented by *mu*, such as *mx*, can be more quickly removed at the lower spindle speed *om*; for *om*.*mx*, which then measures the (time) rate of weight removal, is greater than *om*.*mu*, which would give that rate for machining at the higher spindle speed *on* in the manner above described.

From all of this it follows (still speaking of a job of the same radius) that cuts (proportional to) *mf*, *mx*, *mu*, *mk*, and *mt′*, which, if the spindle were capable of turning at every speed, could be removed at rates represented by *om*.*mf*, *os*.*sw*, *oi*.*iv*, *ol*.*lq*, and *on*.*nt* can actually, owing to the gap between the spindle speeds *om* and *on*, only be removed at rates proportional to *om . mf*, *om . mx*, *om . mu*, *oy . yz*, and *on . nt*.

The ratio of the weight really removable by the lathe to that of which the steel is economically capable is therefore shown by the ratios of the lengths—

$$\frac{ef}{ef'}, \frac{Bx}{Bw}, \frac{ju}{jv}, \frac{rz}{rq}, \text{and } \frac{Ct}{Ct}$$

If all the possible cuts between *mt'* and *mf* are equally likely to occur in the course of a year upon a job of radius $r = \frac{oe}{od}h$, it will appear that the weight actually machined off per annum by a lathe having no intermediate spindle speed between *om* and *on* will bear the same proportion to what the steel is capable of upon a lathe possessing *all* these intermediate speeds which the area *ontzuxfeo* bears to the area *ontqvwfeo*.

Consider next what happens when working upon a job of some smaller radius $r = \frac{\mathrm{OE}}{od}h$, say; for which MF gives the torque for the standard cut a_s.

The tool steel curve must be freshly drawn for the new point F, lying on *bc* vertically above E, which represents this smaller radius; since for the standard area of cut a_s (now represented by MF), the economical cutting speed of the tool (which is the same as before) is now given by the height EF to a different scale.

Let therefore TFHR be the new steel curve drawn to its proper scale for the smaller radius.

The spindle speeds being still *om* and *on* only, but a torque represented by *n*W corresponding to the full power of the lathe being available at the higher speed, weight removal will proceed for cuts from MT' to MF at rates represented by points on the hyperbola TS drawn through T in *n*W, where the steel curve RIHFT cuts the latter.

The weight-removing efficiency of the spindle speed *on*, when working on a radius $r = \frac{\mathrm{OE}}{od}h$, is therefore represented by the ratio of the area under the hyperbola TS to that under the tool steel curve TF.

Cuts greater than a_s, such as MI', could be taken by the steel at a spindle speed $oh = \mathrm{YI}$, if such existed, and if the torque available at this speed were not less than $h\mathrm{I} = \mathrm{MI}'$.

Since this is not the case, the said cut must be removed by going over several times at the given higher speed *on*, and weight removal will only proceed at rates represented as before by the hyperbola TS produced. For all cuts greater than a_s on this radius $\left(r = \frac{\mathrm{OE}}{od}h\right)$, *i.e.* greater than MF, the weight-removing efficiency of the spindle speed *on* is the ratio of the area under the hyperbola TS produced down to vertical *ef* to that under steel curve FIR.

Now, if it be granted that cuts of areas from $\frac{1}{2}a_s$ to a_s are of most frequent occurrence on every diameter of job, and that the weight-removing efficiency of the headstock should be judged with reference to such cuts rather than to cuts greater than a_s [such as those last referred to (MF to MI')], then the triangle-shaped areas TFS and *tf*U show the losses incurred when such cuts are taken by means of spindle speed *on* on jobs of radii $\frac{oE}{od}h$ and $\frac{oe}{od}h$ respectively, and the ratios of $\frac{\text{area } on\text{TSE}}{\text{area } on\text{TFE}}$ and of $\frac{\text{area } ont\text{U}eo}{\text{area } ontfeo}$ show the weight-removing efficiencies of that speed on those jobs.

If all the possible cuts between $\frac{a_s}{2}$ and a_s upon any of the diameters of job between $\frac{n\text{W}}{od}h$ and $\frac{mf}{od}h$ are equally likely to occur in the course of the year, the general result is that the average weight efficiency of spindle speed *on* will be measured by the area under that curve drawn from T to U which is the locus of S, as MF passes from *n*W to *mf*, and the weight-removing efficiency of the speed *on* will be the ratio $\frac{on\text{WSU}e}{on\text{WF}fe}$.

This result must, however, be modified in view of the fact that the greater the number of times the surface of a job has to be machined over—that is, the oftener the saddle has to be returned along the bed, and the more frequently the tool must be reset to a fresh cut or different traverse by the turner—the greater must be the waste of time both by him and by the machine.

In Fig. 243 the cut *n*W (*i.e.* a_s upon $v = \frac{n\text{W}}{od}h$) can be taken without any loss, because the steel is capable of it at the spindle speed *on*, and because this spindle speed and the torque required to take a_s on the given diameter of job are both available. The cut MF (*i.e.* a_s on $r = \frac{\text{MF}}{od}h$), however, has to be taken twice over (in portions each equal to MT′ at spindle speed *on*; and the rate of weight removal, viz. $on \,.\, n\text{T}$ ($= \text{OE} \,.\, \text{ES}$), indicated by the point S on the hyperbola TS is really too favourable. Similarly, cuts such as *mk* or *mf* (*i.e.* cuts $\frac{mk}{mf}a_s$ and a_s itself on radius $r = \frac{mf}{od}h$), if taken at spindle speed *on*, can only be removed by several successive cuts of area mt' ($= nt$); and the rate proportional to $oy \,.\, yz = oe \,.\, e\text{U}$, which theoretically gives the rate of removal, is really far too high.

We have found that a more reasonable approximation to the rate of weight removal of all such cuts as from $\frac{1}{2}a_s$ to a_s on jobs of radii from $\frac{n\text{W}}{od}h$ to $\frac{mf}{od}h$ is obtained by taking, instead of the curve WSU (locus of S), the straight line WE′ drawn from W which bisects the angle between the vertical through W and the tangent to the hyperbola *bc* passing through W. This line WE′, drawn to cut the next speed, say *mf* in D, will more truly represent by the ratio $\frac{\text{area} = on\text{WD}fe}{\text{area} = on\text{WF}fe}$ the fraction of the weight

$$\frac{\text{removable by the given speeds } on \text{ and } om}{\text{that removable by a lathe with every speed}}$$

The characteristic should obviously leave the straight line WD and go along the horizontal D*f* of the next lower speed; since the rate of weight removal for all cuts such as MF would be increased by taking them at speed *om* rather than *on* (the rate of removal at speed *om* is $o\mathrm{E}.\mathrm{E}j$; but for the higher speed *on* it is OE . EE′).

(*b*) **Effect of Defective Torques.**—If, now, instead of a torque equal to *n*W, one equal to *nt* only is provided at spindle speed *on*, we see at once that the weight removable is no longer limited by the capacity of the tool steel, but by the pulling power of the belt. With any cut from *mt′* to *mf*, or from M*t″* to MF, weight removal at speed *on* can take place at no greater rate than $on . nt$ (where *nt* is the actually provided torque at revolutions *on*).

Thus, if there were no loss due to sliding back and resetting when a given area of cut has to be machined in several successive cuts of smaller area, the ratio $\dfrac{\text{weight removable at speed } on \text{ with torque } nt}{\text{weight removable with every spindle speed provided}}$ would be—

$$\frac{ont\mathrm{U}e}{on\mathrm{W}fe}$$

It would be, here again, more economical to remove cuts greater than *mu* at the lower speed *om*, and the correct efficiency will therefore be given by the ratio $\dfrac{ontufe}{on\mathrm{W}fe}$.

Allowing for loss of time due to sliding back and resetting, the efficiency ratio becomes—

$$\zeta = \frac{ont\varepsilon\mathrm{D}f\varepsilon}{on\mathrm{W}fe}$$

Finally, if a torque equal only to *mt′* had been provided at the lower speed *om*, the rate of weight removal at that speed could not exceed $om . mt'$. The point δ where the hyperbola through t' cuts the bisector WDδE′ would indicate the cut for which an equal rate of weight removal would then be given by either of the spindle speeds (*on* or *om*). For cuts of greater area the lower speed would give better results, and the weight removal characteristic curve should thereafter follow the hyperbola $\delta\lambda$.

Thus we obtain the law for drawing the characteristic—

Law.—*The bisecting line (above described) drawn from the intersection of any spindle-speed-line produced to cut the power curve (bc) becomes part of the characteristic whenever it forms the intercept between the hyperbolas through the torque extremities of that spindle speed and the next below. If the torque lines extend beyond the bisector, then the bisecting line forms part of the characteristic between those torque lines themselves.*

The method of drawing the characteristic will, however, be best understood by reference to the practical examples given below.

The method of studying headstock efficiencies here developed (but by no means exhaustively treated) seems to offer advantages which should render the question of its adoption by lathe designers at least worthy of inquiry. It may serve similar purposes for them which the combined indicator diagram fulfils for the steam engineer.

CHAPTER XLII

APPLICATION OF THE TORQUE-SPEED DIAGRAM AND THE LATHE CHARACTERISTIC TO EXAMPLES FROM PRACTICE

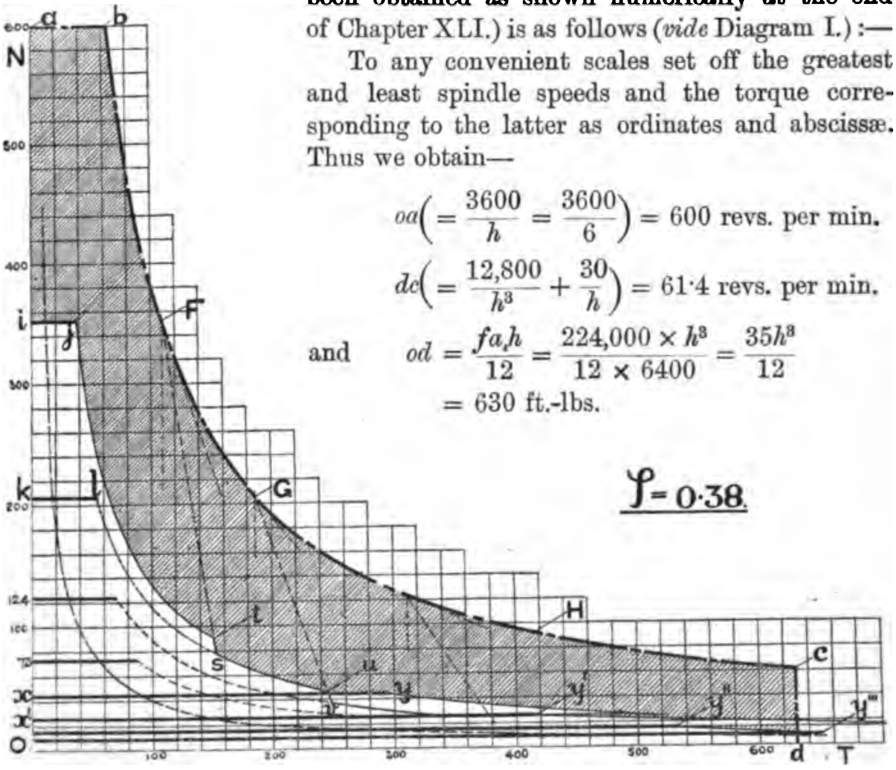
Example I.—The first example of the application of our method to the determination of the economic efficiency of a headstock is that already referred to at the end of Chapter XL. The method of plotting the characteristic for the given lathe (whose torque-speed lines *ij.bloz* have been obtained as shown numerically at the end of Chapter XLI.) is as follows (*vide* Diagram I.):—

To any convenient scales set off the greatest and least spindle speeds and the torque corresponding to the latter as ordinates and abscissæ. Thus we obtain—

$$oa\left(=\frac{3600}{h}=\frac{3600}{6}\right)=600 \text{ revs. per min.}$$

$$dc\left(=\frac{12,800}{h^3}+\frac{30}{h}\right)=61\cdot4 \text{ revs. per min.}$$

and
$$od=\frac{fa_ih}{12}=\frac{224,000\times h^3}{12\times 6400}=\frac{35h^3}{12}$$
$$=630 \text{ ft.-lbs.}$$



Torque Speed Diagram for 6-in Lathe (cone on spindle).

Diagram I.

Then *b* and *c* are points on a rectangular hyperbola of constant power which we take as the standard of reference, or the *characteristic of a perfect lathe.*

$$\left.\begin{array}{rl} \text{Next set off } ij = & 39 \text{ ft.-lbs.} \\ \text{at spindle revolutions } oi = & 352 \quad ,, \\ \text{Also } kl = & 55 \quad ,, \\ \text{at spindle revolutions } ok = & 206 \quad ,, \end{array}\right\} \begin{array}{c} \text{as found in Chapter XL.,} \\ \text{p. 352 } \textit{ante.} \end{array}$$

and so on for every spindle speed the lathe possesses.

Place the diagram (which is preferably upon tracing-paper) upon the $(N - r)$ chart so as to enable hyperbolas to be drawn where required with facility. Thus the hyperbola of standard power *bc*, also those through the ends of the torque lines for each spindle speed, can easily be obtained (e.g. *jt*, *ls*, etc.).

Through the points F, G, etc., where the spindle speeds provided cut the *bc* line, draw straight lines F*t*, G*u*, etc. (bisectors of the angles between verticals and tangents through F and G) in the manner above indicated. F*t* must form the intercept *ts*, and *Gu* the intercept *uv* between the two hyperbolas drawn from its own and the next lower spindle speed torque lines.

In the present instance the line G*u* cuts the torque line *xy* nearly in *u*. This means that more weight can be removed by making use of speed *ox* rather than any of the higher speeds with which the headstock is provided, if the job be of greater radius than $\frac{xu}{od}h$.

The fact that the hyperbola through *y* (carried out to the vertical *cd*) passes above all subsequent hyperbolas, such as those through y', y'', and y''', shows that a greater rate of weight removal can be effected by using the spindle speed *ox* rather than any of the lower ones with which the lathe is fitted.

In fact, the spindle-speeds 124, 73, 26, 15·5, and 9 might be entirely dispensed with, without any diminution of the weight-removing efficiency of the headstock so far as high-speed steel is concerned. Speeds 26, 15·5, and 9 will, however, be of use for work with carbon steel, or when cutting very hard materials; but speeds 124 and 73 are apparently quite valueless.

The area shown shaded in Diagram I. measures the inefficiency of this lathe relatively to our high-speed steel standard *oabcd*. The ratio of the unshaded to the latter area, which we have called the weight-removing efficiency, gives a good estimate of the value of the headstock as an economical cutting machine relatively to one having a continuously variable speed revolution sequence, and capable of giving such a torque as each speed as corresponds to the standard horse-power we have postulated. This ratio, which we have denoted by ζ, has in this instance the value $\zeta = 0{\cdot}38$, and by referring to Fig. 241, *ante*, we can at once set down the increased cost which is incurred in the removal of one ton of steel by the use of the defective design of headstock now under examination.

The cost of machining one ton upon a "perfect" 6-in. lathe (one, that is, for which $\zeta = 1$) is by that figure 213 pence, whilst for this headstock for which $\zeta = 0{\cdot}38$, the cost is 365 pence per ton. This means an increase from 12*s*. 8*d*. to 17*s*. 9*d*. per ton, or over 71 per cent. increase upon the minimum cost.

This, it may be remarked, is by no means an exceptional case. It is a type of lathe in very common use, made by a first-class manufacturer, and sold by him as being a suitable machine for use with high-speed steel.

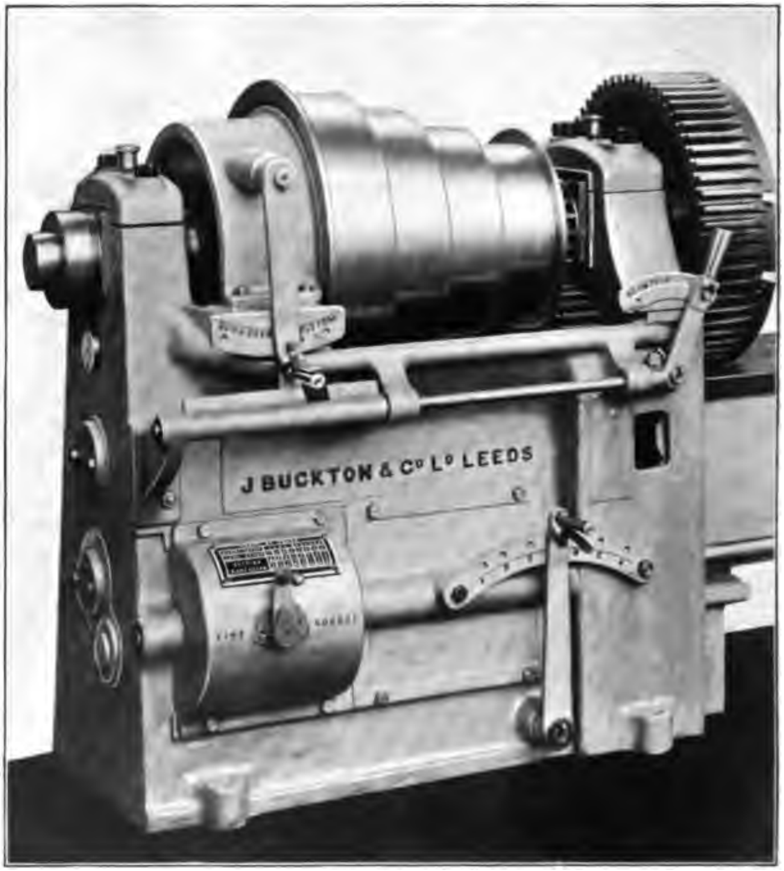
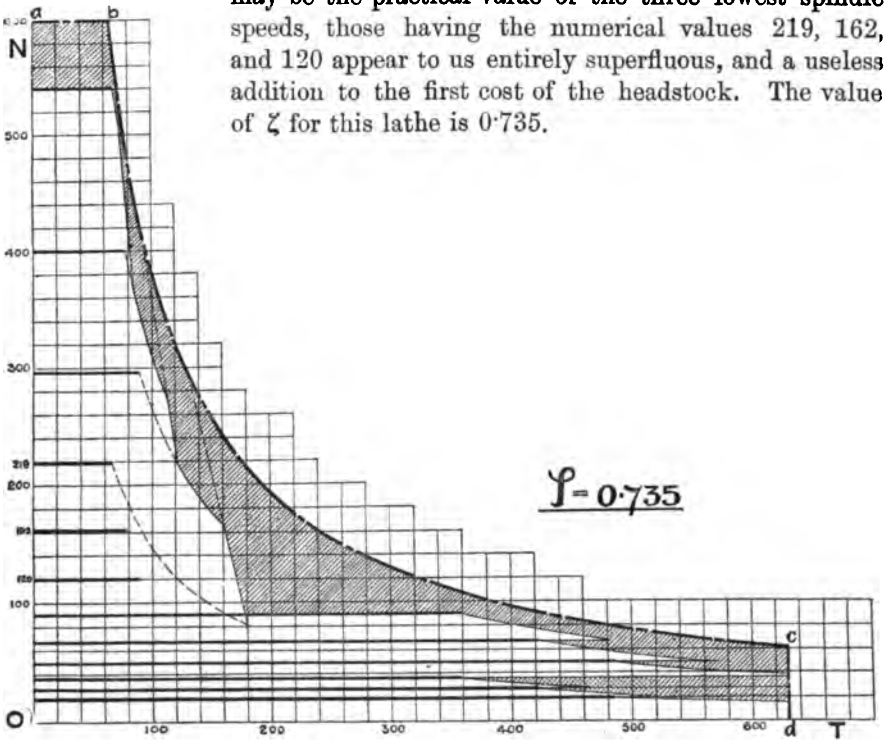


Fig. 244.—Headstock for 12 to 14-in. Centres Lathes (Buckton).

Example II.: 6-in. Centres Lathe, Cone on Spindle.—Diagram II. shows another example of a 6-in. lathe treated in exactly the same way. Whatever may be the practical value of the three lowest spindle speeds, those having the numerical values 219, 162, and 120 appear to us entirely superfluous, and a useless addition to the first cost of the headstock. The value of ζ for this lathe is 0·735.



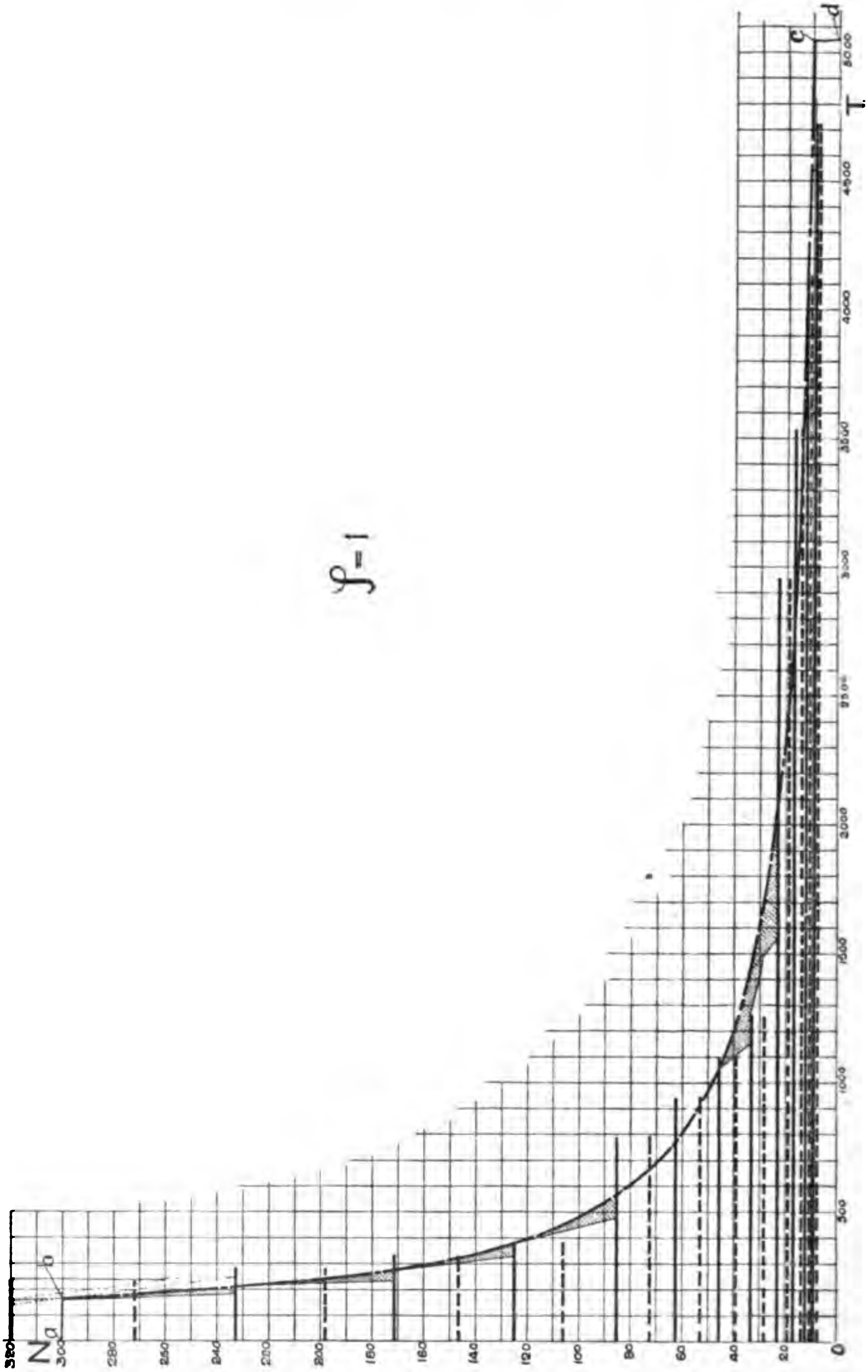
Torque Speed Diagram for 6-in. Lathe.

Diagram II.

Its cost for machining one ton is therefore 250 pence instead of 213 pence; this corresponds to 3*s.* 1*d.*, or 17½ per cent. additional to the proper charge upon a lathe of this size.

Example III.: 12-in. Centres Lathe, Cone on Spindle (Fig. 244, and Diag. III.).—The headstock for the 12-in. centre lathe by J. Buckton & Co. is shown by Fig. 244. In this lathe, which has been specially designed for taking heavy cuts and feeds at a high speed, the headstock is continued down to the floor, and forms a box casting containing and protecting all the driving gear and feed gear. The spindle is of steel, with front neck 5½ in. diameter by 7½ in. long, running in gun-metal bearings. The belt cone has four steps of diameters 24 in., 21 in., 18 in., and 15 in., and width of face suitable for a 5-in. belt. There are two changes of gearing: a ring of gear upon the face-plate giving ratios of 3·75 and 14 to 1, and a total range of twelve speeds uniformly graded in geometrical progression over the whole range; the changes of gear being effected by handles conveniently situated on the front of the headstock and interlocked. There are eight changes of feed, operated by positive gearing arranged in uniform steps from $\frac{1}{32}$ in. to ¼ in. These can be easily changed without stopping the lathe by a handle conveniently placed at the front of the headstock.

The saddle and slide rests for this machine were described on p. 262, *ante*, and are illustrated by Fig. 161, at the end of the book.



Torque Speed Diagram for 12-in. Lathe (cone on spindle).

DIAGRAM III.

The loose headstock is provided with a steel spindle of 3¼ in. diameter, having adjustment by screw and hand-wheel, and can be locked in any position.

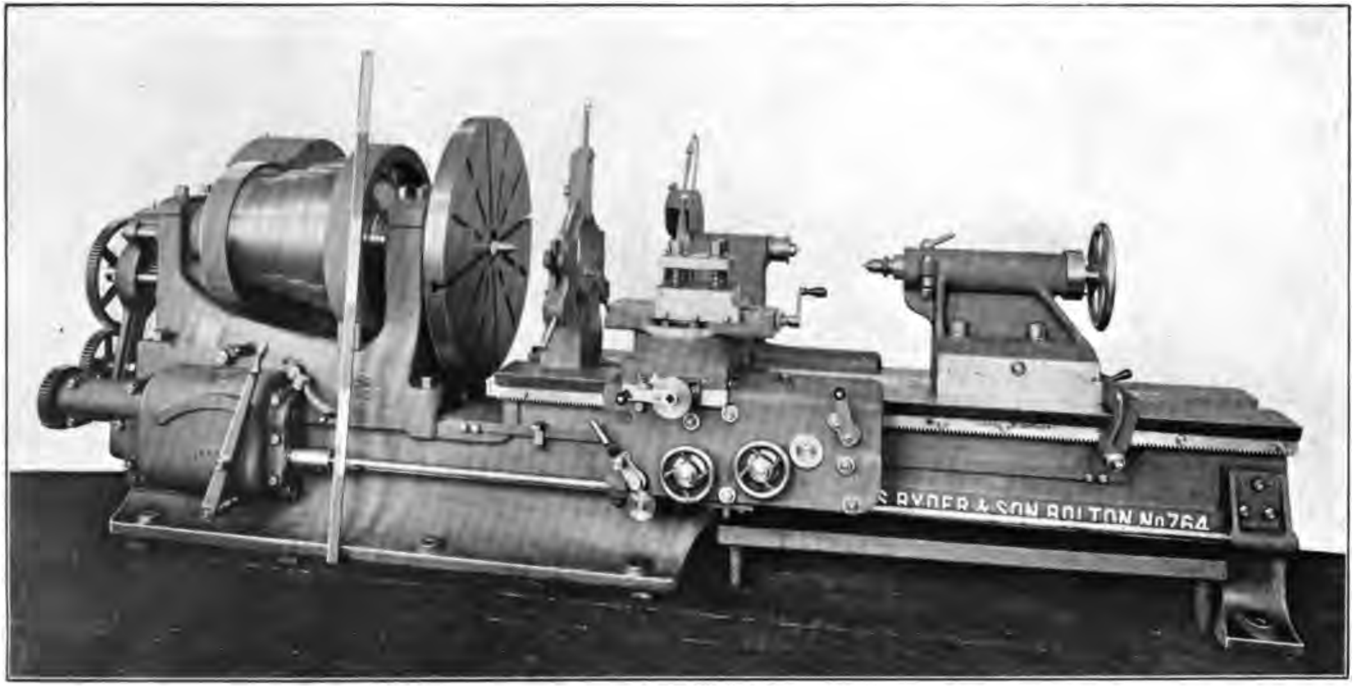


FIG. 245.—CONE-DRIVEN STANDARD 13-IN. CENTRES HIGH-SPEED LATHE (RYDER).

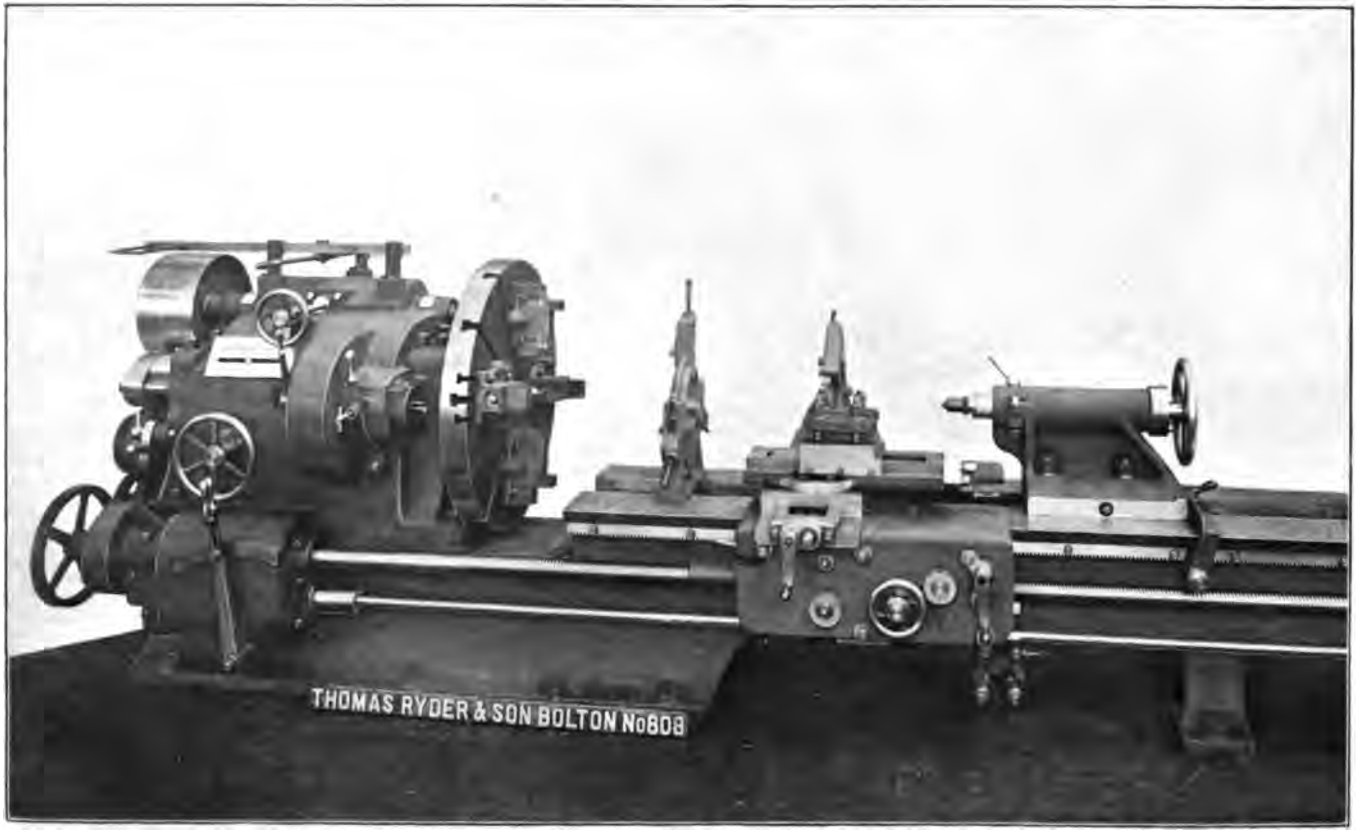


Fig. 246.—All-gear Head Standard 13-in. Centres High-speed Lathe (Ryder).

Two counter-motion speeds may be supplied with the machine. For this size they are 200 and 170 revolutions per minute.

On Diagram III. we have shown the power curve for a 12-in. lathe, and have plotted the torques and spindle speeds as deduced from the specifications for a 12-in. and for a 14-in. lathe supplied to us by the makers. They are tabulated below.

We have made no deduction for journal friction or other loss when running direct; but for the back-gear spindle speeds we have assumed that the mechanical efficiency of each pair of gears is 0·95. The annexed schedule gives the speeds and torques for the higher countershaft speed only. Those for the lower speed are shown by the dotted lines of Diagram III.

$$\begin{array}{llll}
N_1 = \frac{200 \times 24}{15} = 320 & T_1 = \left(\frac{5 \times 75}{24}\right) & \times \quad 15 & = 234 \\
N_2 = \frac{200 \times 21}{18} = 233 & T_2 = \quad ,, & \times \quad 18 & = 280 \\
N_3 = \frac{200 \times 18}{21} = 171 & T_3 = \quad ,, & \times \quad 21 & = 327 \\
N_4 = \frac{200 \times 15}{24} = 125 & T_4 = \quad ,, & \times \quad 24 & = 374
\end{array}$$

$$\begin{array}{ll}
N_5 = \frac{1}{3{\cdot}75} \times 320 = 85 & T_5 = \frac{5 \times 75}{24} \times 15 \times 3{\cdot}75 \times (0{\cdot}95)^2 = 790 \\
N_6 = \ ,, \times 233 = 62 & T_6 = \ ,, \quad \times 3{\cdot}75 \times (0{\cdot}95)^2 \times 18 = 944 \\
N_7 = \ ,, \times 171 = 46 & T_7 = \ ,, \quad ,, \quad ,, \quad \times 21 = 1100 \\
N_8 = \ ,, \times 125 = 33 & T_8 = \ ,, \quad ,, \quad ,, \quad \times 24 = 1260
\end{array}$$

$$\begin{array}{ll}
N_9 = \frac{1}{14} \times 320 = 23 & T_9 = \frac{5 \times 75}{24} \times 14 \times (0{\cdot}95)^2 \times 15 = 2950 \\
N_{10} = \ ,, \times 233 = 17 & T_{10} = \ ,, \quad ,, \quad ,, \quad \times 18 = 3580 \\
N_{11} = \ ,, \times 171 = 12 & T_{11} = \ ,, \quad ,, \quad ,, \quad \times 21 = 4130 \\
N_{12} = \ ,, \times 125 = 9 & T_{12} = \ ,, \quad ,, \quad ,, \quad \times 24 = 4720
\end{array}$$

An inspection of Diagram III., in which these values of N and T have been plotted, shows that they are well up to, and in many cases in excess of, the values required by our standard torque-speed diagram for a 12-in. lathe. It ought to be mentioned, in explanation, that the makers supply the same identical headstock for both their 14-in. and 12-in. centres lathe, and that this headstock might more properly have been compared with our 14-in. torque-speed power curve.

The "bisectors" have been drawn as indicated; but as the hyperbolas through the ends of the actual torque lines lie to the right of them, these bisectors are in this case terminated by the torque horizontals themselves. The shaded areas show the losses incurred by the gaps in the continuity of the spindle speeds; but allowing for the torques in excess of our power curve, the efficiency of the characteristic for this lathe may be taken at unity. The cost per ton of steel machined (assuming that this lathe costs no more than £210, as per our formula, $C = 5h^{\frac{3}{2}}$) is, according to Fig. 241 (12-in. lathe, $\zeta = 1$), 275*d*. or 23*s*. per ton.

The salient feature of this lathe characteristic has reference to the spindle speeds (shown dotted in Diagram III.) provided when the second or lower countershaft speed is supplied. Our diagram proves that cuts taken with those spindle speeds could in every case be removed more economically by using either the next higher or the next lower (spindle speed) obtainable from the single or higher counter-motion speed alone. The addition of the second counter-motion speed, though ostensibly doubling the number of spindle speeds available, is of no advantage whatever, and, in reality, detracts from the efficiency of the headstock by enabling the turner to run at cutting speeds below the proper capacity of the steel.

Examples IV. and V.—**13-in. Centres Lathes, by Messrs. T. Ryder and Sons.** *Example* 4, *Cone on spindle* (Fig. 245 and Diag. IV.); *Example* 5, *All-gear head* (Figs. 246, 247, 248 and Diag. V.).

In lathes from 8 to 16-in. centres, Messrs. Ryder mount the cone on the spindle, but step the bed in order that the diameter of the largest cone step may at least equal 2*h*. The diameter of the bottom step is only 25 per cent. less than the top step, thus giving practically a uniform tension of strap throughout the system, and permitting the smaller step to retain its place in the system.

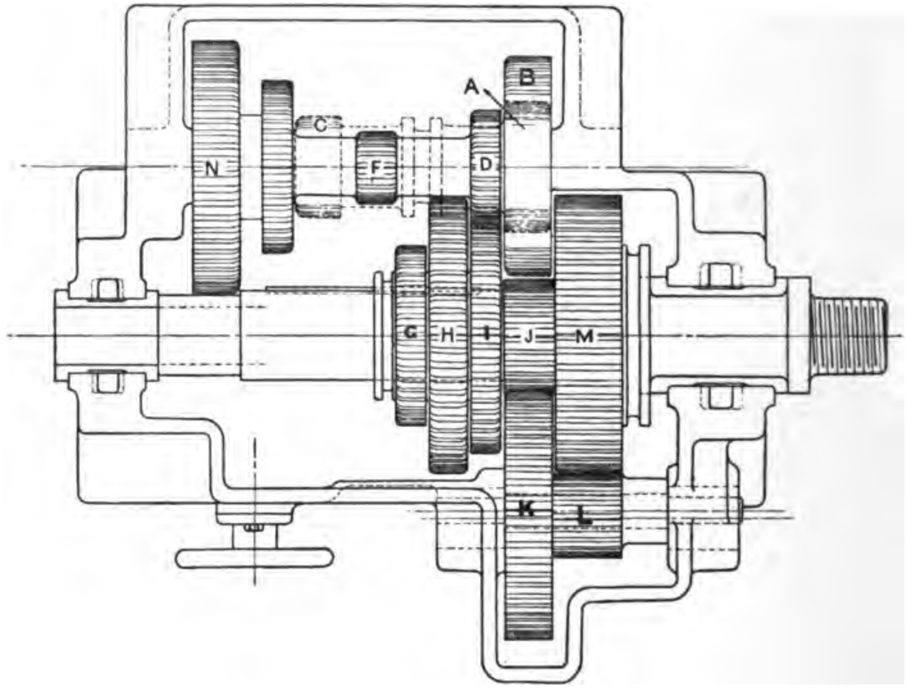


Fig. 247.—Arrangement of Gear for 13-in. Headstock.

These machines are fitted with two countershaft speeds and two sets of back gears, which give 24 changes of speeds, with a ratio of 80 to 1 to the spindle. The back-gear wheels are put in or out of action by a handle conveniently placed at the front of the headstock, which operates the levers on the eccentric bushes in the headstock cheeks; whilst a second handle, also at front of the headstock, engages either one or other of the sets as shown in Fig. 245 (at the end of the book), which represents the 13-in. lathe. This machine has an emery grinding head fitted to the back of the saddle, which is actuated transversely by a handle at the front of the saddle, and is driven from a drum on the countershaft. The saddle is traversed along the bed by the usual feed mechanism, whilst the reversing gear in the saddle is actuated by the handle at the lower left-hand side of the saddle apron. The rectangular rod fitted to the bed above the feed shaft is provided with trip dogs which act as an automatic "knock out" to the feed mechanism, or a reverse for grinding. These machines are, in our opinion, characterized by the attention which in their design has been given not only to strength, rigidity, and power, but also to handiness and convenience of operation. The principal dimensions of four sizes of cone-driven lathes are given in the table on p. 372.

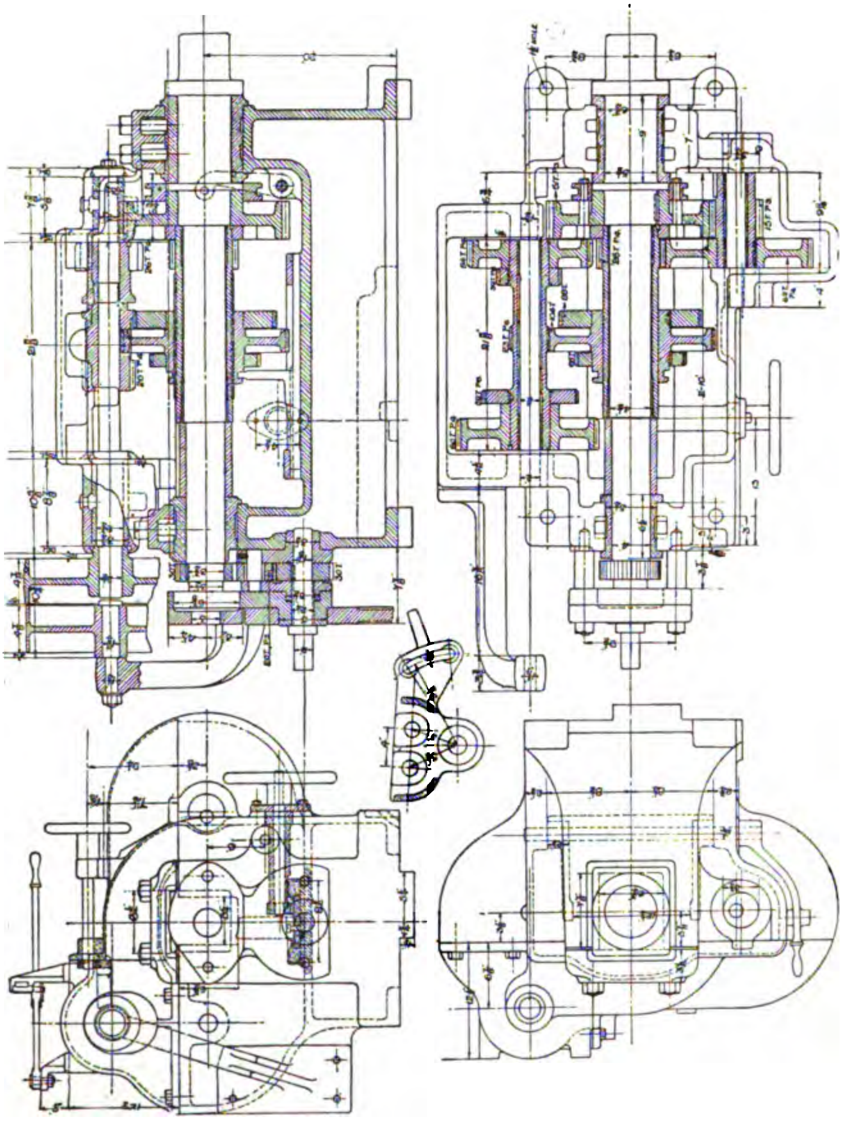


FIG. 248.—20-IN. HEADSTOCK FOR 18-IN. LATHE. ALL-GEAR HEAD.

A general view of an 18-in. centres high-speed lathe with cone mounted on a side shaft is shown by Fig. 165, at the end of the volume, and the following describes it in detail.

The fast headstock is provided with a five-stepped cone, and one single and two treble gears. The arrangement of the cone and gears is similar to that shown in Fig. 53, p. 114, but with an internal faceplate wheel. The gear ratios are as follows: Single gear, 6 to 1; treble gears, 22 and 85 to 1.

PARTICULARS OF THOMAS RYDER AND SONS' "MASSIVE" HIGH-SPEED CUTTING LATHES.

Height of centres . .	8 in.	10 in.	13 in.	16 in.
Bed, width	14½ in.	18 in.	23½ in.	29 in.
Proportion of gearing in the fast headstock	12⅓ to 1	19½ to 1, and 4½ to 1	21½ to 1, and 4½ to 1	24½ to 1, and 5 to 1
Cone diameters . . .	4 speeds, 17½ to $10\frac{11}{16}$ in.	4 speeds, 21 to $15\frac{15}{16}$ in.	4 speeds, 26 to 19⅝ in.	4 speeds, 34 to 25 in.
Width of belt on cone	3¼ in.	3½ in.	4½ in.	5 in.
Front bearing of spindle	3½ × 6 in.	4¼ × 7¼ in.	5½ × 9 in.	6¾ × 11 in.
Rear bearing of spindle	2½ × 4¾ in.	3 × 5½ in.	4 × 6½ in.	5 × 8¼ in.
Depth of cut × traverse	$\frac{3}{64} \times \frac{1}{16}$ in.	$\frac{13}{64} \times \frac{3}{32}$ in.	$\frac{1}{4} \times \frac{7}{64}$ in.	$\frac{5}{16} \times \frac{9}{64}$ in.
Diameter of loose headstock spindle .	2½ in.	3 in.	3½ in.	4 in.
Lathe will swing over carriage	11¼ in.	14¼ in.	19¼ in.	24 in.
Will swing in opening without gap-piece .	24 × 7 in.	30 × 9 in.	40 × 12 in.	48 × 15 in.
Length of bed . . .	8 ft.	10 ft.	13 ft. 9 in.	15 ft. 3 in.
Will admit between centres	3 ft. 4½ in.	4 ft. 4 in.	6 ft. 6 in.	6 ft. 9 in.
Size of counter pulleys	18-in. diameter for 3½-in. belt	20-in. diameter for 4-in. belt	24-in. diameter for 4½-in. belt	32-in. diameter for 5½-in. belt
Approximate nett weight	42 cwts.	78 cwts.	145 cwts.	200 cwts.

A two-speed countershaft is provided, thereby giving thirty changes of speeds to the spindle. The diameter of the top and bottom cone steps is 30 and 20 in. respectively, and all are wide enough to receive a 5-in. belt.

The cone shaft and main spindle run in phosphor bronze, adjustable, capped bearings. The spindle, which is of high carbon, hammered steel, has a front neck 8 in. diameter by 11 in. long, and the rear bearing is 5 in. diameter by 8¼ in. long. The end thrust is taken by ball bearings. The four-change feed to the saddle is placed at the back of the lathe, but controlled from the front, and either or both of the saddles can be engaged by suitable mechanism on each.

The saddles are mounted on a three-way bed, similar to the 16-in. bed shown in Fig. 216, and is 29 in. over the ways and 19 in. deep. The front way is made specially wide to provide a support to the saddle when operating on work of large diameter. The guide-screw is of steel, 3½ in. diameter by ½ in. pitch, and engages with the saddles by means of double-clasp nuts. The front wing of the saddle is extra long, and a solid square lip running its full length bears on the inside vertical face of the front way of the bed, whilst a similar lip and adjustable tapered strip bearing on the outer vertical face preserves the lateral alignment of the carriage. Two long adjustable strips fitted to the saddle, and bearing on the underside of back and front lip of the bed, hold the saddle in position.

The rear wing of the saddle is cut away to allow of the loose headstock being brought close to the work without undue overhang of the spindle. The loose headstock spindle is 4½ in. diameter, and both the barrel and base are of extra length. The fast and loose headstock centres are made to No. 6 Morse taper.

The machine throughout is of substantial proportions, and made to take 7 ft. between centres, with a bed 15 ft. 6 in. long, to swing 28 in. diameter over the saddles, and 50 in. diameter by 15½ in. long in the gap.

Figs. 246, 247, and 248 show the 13-in. centre all-gear head lathe, by this firm. It is arranged to be driven direct from the line shaft, or if more than one speed to the pulley is desired a countershaft is provided.

The arrangement of the gear is shown in Fig. 247. The driving pulley is keyed to the shaft carrying the pinions A and C, which are also keyed to a sleeve, and free to slide on the shaft. These two pinions are made to gear with the wheels B and N on the second motion shaft, situated immediately below the pulley shaft, by means of a claw, rack, and pinion actuated by a hand wheel on the front of the headstock. The wheels E, F, and D are also keyed to the second-motion shaft, and respectively mesh with the wheels G, H, and I, keyed to a sleeve on the spindle, but free to slide on the same by means of mechanism similar to the above. The pinion J is also keyed to this sleeve. The wheel M and clutch are coupled together and keyed to the spindle, the clutch being free to slide forward and engage with J when running single gear. K and L are keyed to a sleeve revolving freely on an eccentric shaft, and engaged with J and M by means of the handle shown at the front of the headstock.

The gear is so arranged that it is impossible to have more than one set of gears engaged at one time, and the double gear cannot be engaged until the clutch has been withdrawn.

The positions of the hand wheels and handles for these various motions are clearly shown in the figures, together with dials and fingers which indicate the revolutions to the spindle for the particular gears engaged. All these gears are of mild steel and machine-cut.

The gears are as follows:—

Double gears.	Quadruple gears.
A, B, E, G	A, B, E, G, J, K, L, M
C, N, E, G	C, N, E, G, J, K, L, M
A, B, D, I	A, B, D, I, J, K, L, M
C, N, D, I	C, N, D, I, J, K, L, M
A, B, F, H	A, B, F, H, J, K, L, M
C, N, F, H	C, N, F, H, J, K, L, M

The reversing wheels (Fig. 248) are of steel, and machine-cut, and are large in diameter. They are bushed with brass, and the shafts carrying the intermediate wheels are supported at each end by the reversing frame. The last wheel in the reversing train gears with a wheel keyed to a loose sleeve on the guide screw, whereon are also keyed four other wheels which mesh with four others on the feed shaft. Any one of these can be engaged by a sliding cotter, which is operated by a lever at the front of the lathe.

Independent mechanism is provided in the carriage apron for the self-acting surfacing and sliding feeds, and for a fine and coarse hand traverse motion. A double clasp nut engages the steel guide screw. The bed is of complete box section, and is of a design eminently suitable for resisting torsional stress. The top and bottom flanges within the ways extend the full length of the bed, and pass-holes for cuttings and lubricant are provided at intervals. The bed is stayed transversely by the usual box bars. The front surface is very broad, and the headstock centres are set back from the centre of the opening in the bed, to allow of a rigid support to the tool when operating on large diameters of work. A rack is cast within the ways to receive a chock fitted to the loose headstock for taking the backward thrust of this head. The loose headstock is arranged for taper turning, and has a spindle 3 in. diameter, which is provided with a self-acting boring feed. This feed is obtained from the feed shaft at the front transmitting its motion through the wheels at the loose end of the bed to a shaft at the rear, whereon is mounted a worm. This slides with the head, and gears with a worm wheel on a vertical shaft. On this last shaft is a mitre which gears with another on the shaft behind the head carrying the spur pinion which meshes with the spur wheel on the spindle screw. A clutch at the end of this screw rapidly engages or disengages the feed. The whole machine is neat, compact, and convenient to handle.

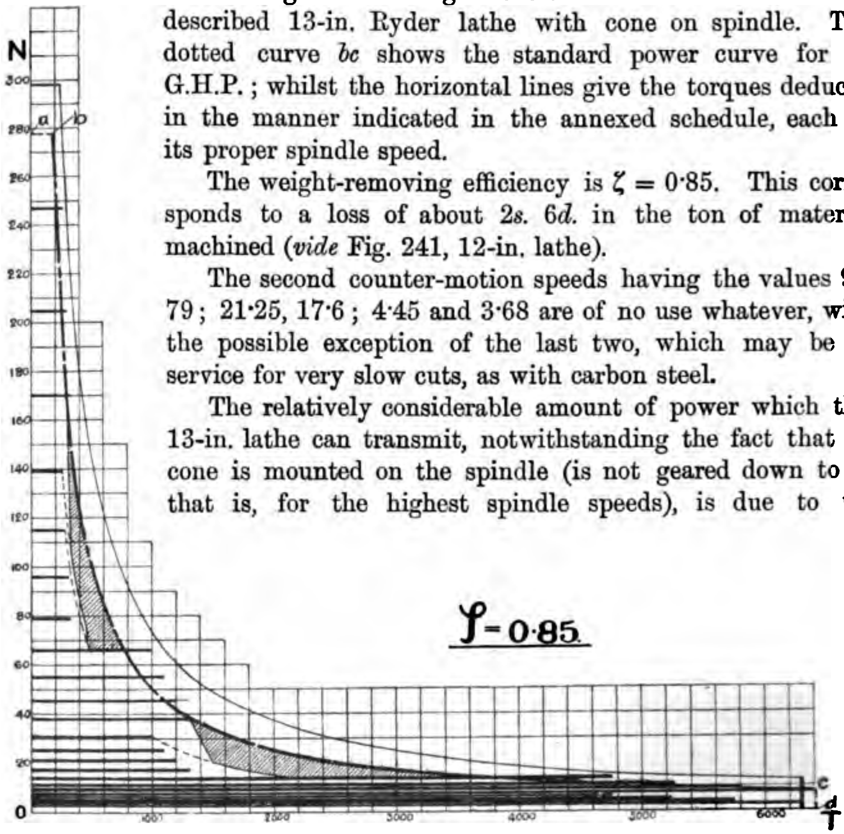
The same lathe is supplied with an electric motor drive when required.

In Diagram IV. we give the characteristic for the above-described 13-in. Ryder lathe with cone on spindle. The dotted curve *bc* shows the standard power curve for 11 G.H.P.; whilst the horizontal lines give the torques deduced in the manner indicated in the annexed schedule, each at its proper spindle speed.

The weight-removing efficiency is $\zeta = 0{\cdot}85$. This corresponds to a loss of about 2*s.* 6*d.* in the ton of material machined (*vide* Fig. 241, 12-in. lathe).

The second counter-motion speeds having the values 96, 79; 21·25, 17·6; 4·45 and 3·68 are of no use whatever, with the possible exception of the last two, which may be of service for very slow cuts, as with carbon steel.

The relatively considerable amount of power which this 13-in. lathe can transmit, notwithstanding the fact that its cone is mounted on the spindle (is not geared down to it, that is, for the highest spindle speeds), is due to the stepping down of the bed under the first headstock. This enables much



Torque Speed Diagram for 13-in. Lathe (cone on spindle).

DIAGRAM IV.

larger diameters of cone step to be used than ordinarily; so increasing not only the belt speed, but the effective tension which any given belt can transmit per inch of width.

13-in. Lathe. Cone on Spindle.

Cone step diameters, 26, 23⅞, 21¾, 19⅝ × 4¼ wide.
Countershaft speeds, 105 and 225.
Belt width taken at 4 in.; *effective tension*, 75 lbs. per inch width.
Gear ratios, 4·5 and 21·5 to 1.

$$N_1 = \frac{225 \times 26}{19{\cdot}625} = 298 \qquad T_1 = \left(\frac{4 \times 75}{24}\right) \times 19{\cdot}625 = 245$$

$$N_2 = \frac{225 \times 23{\cdot}875}{21{\cdot}75} = 247 \qquad T_2 = \text{,,} \times 21{\cdot}75 = 272$$

$$N_3 = \frac{225 \times 21{\cdot}75}{23{\cdot}875} = 205 \qquad T_3 = \text{,,} \times 23{\cdot}875 = 298$$

$$N_4 = \frac{225 \times 19{\cdot}625}{26} = 170 \qquad T_4 = \text{,,} \times 26 = 325$$

$$N_5 = \frac{105 \times 26}{19{\cdot}625} = 139 \qquad T_5 = \text{,,} \times 19{\cdot}625 = 245$$

$$N_6 = \frac{105 \times 23{\cdot}875}{21{\cdot}75} = 115 \qquad T_6 = \text{,,} \times 21{\cdot}75 = 272$$

$$N_7 = \frac{105 \times 21{\cdot}75}{23{\cdot}875} = 95{\cdot}7 \qquad T_7 = \text{,,} \times 23{\cdot}876 = 298$$

$$N_8 = \frac{105 \times 19{\cdot}625}{26} = 79{\cdot}3 \qquad T_8 = \text{,,} \times 26 = 325$$

$$N_9 = \frac{1}{4{\cdot}5} \times 298 = 66{\cdot}3 \qquad T_9 = \left\{\frac{4 \times 75}{24} \times 4{\cdot}5 \times (0{\cdot}95)^2\right\} \times 19{\cdot}625 = 992$$

$$N_{10} = \text{,,} \times 247 = 54{\cdot}9 \qquad T_{10} = \text{,,} \quad \text{,,} \times 21{\cdot}75 = 1100$$

$$N_{11} = \text{,,} \times 205 = 45{\cdot}6 \qquad T_{11} = \text{,,} \quad \text{,,} \times 23{\cdot}875 = 1206$$

$$N_{12} = \text{,,} \times 170 = 37{\cdot}8 \qquad T_{12} = \text{,,} \quad \text{,,} \times 26 = 1315$$

$$N_{13} = \text{,,} \times 139 = 30{\cdot}9 \qquad T_{13} = \text{,,} \quad \text{,,} \times 19{\cdot}625 = 992$$

$$N_{14} = \text{,,} \times 115 = 25{\cdot}6 \qquad T_{14} = \text{,,} \quad \text{,,} \times 21{\cdot}75 = 1100$$

$$N_{15} = \text{,,} \times 95{\cdot}7 = 21{\cdot}25 \qquad T_{15} = \text{,,} \quad \text{,,} \times 23{\cdot}875 = 1206$$

$$N_{16} = \text{,,} \times 79{\cdot}3 = 17{\cdot}61 \qquad T_{16} = \text{,,} \quad \text{,,} \times 26 = 1315$$

$$N_{17} = \frac{1}{21{\cdot}5} \times 298 = 13{\cdot}88 \qquad T_{17} = \left\{\frac{4 \times 75}{24} \times 21{\cdot}5 \times (0{\cdot}95)^2\right\} \times 19{\cdot}625 = 4750$$

$$N_{18} = \text{,,} \times 247 = 11{\cdot}5 \qquad T_{18} = \text{,,} \quad \text{,,} \times 21{\cdot}75 = 5260$$

$$N_{19} = \text{,,} \times 205 = 9{\cdot}55 \qquad T_{19} = \text{,,} \quad \text{,,} \times 23{\cdot}875 = 5760$$

$$N_{20} = \text{,,} \times 170 = 7{\cdot}58 \qquad T_{20} = \text{,,} \quad \text{,,} \times 26 = 6290$$

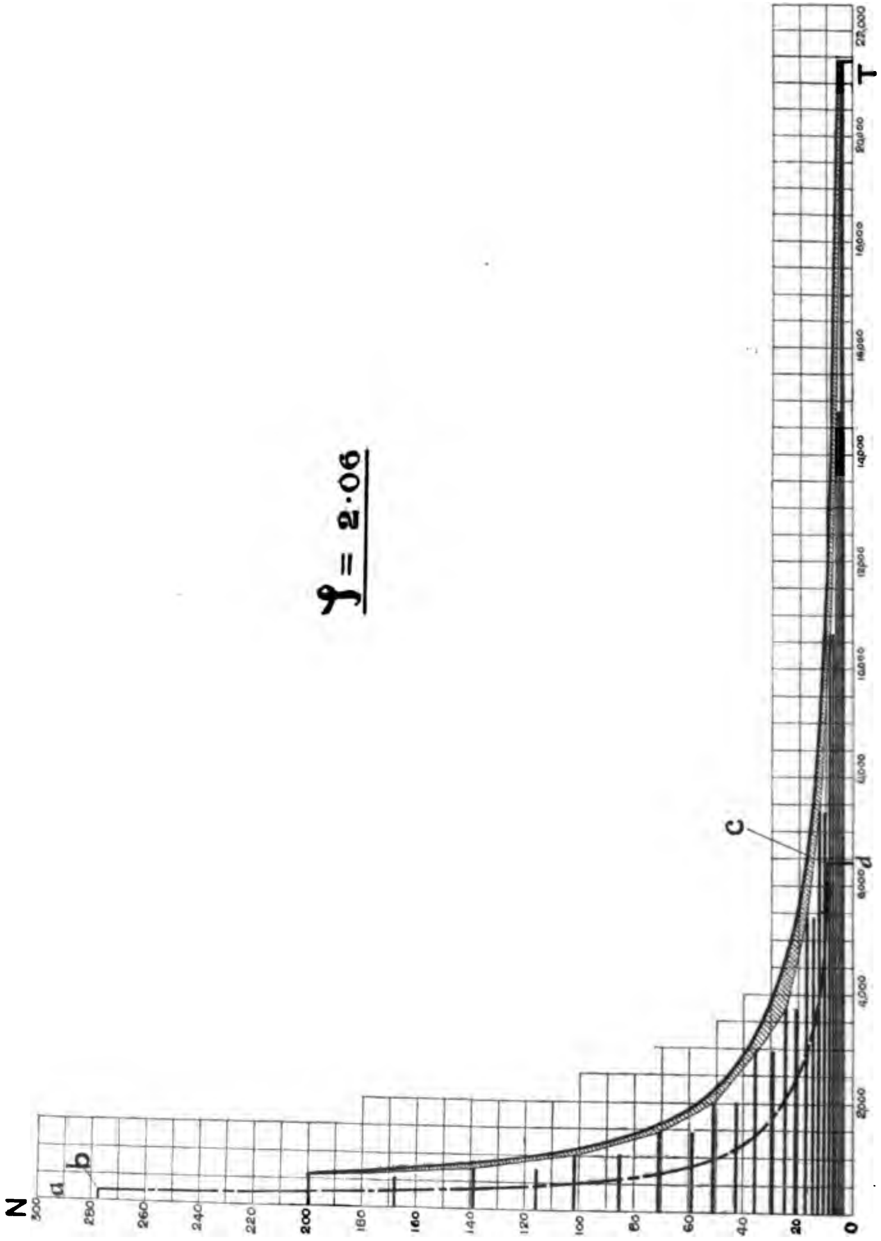
$$N_{21} = \text{,,} \times 139 = 6{\cdot}46 \qquad T_{21} = \text{,,} \quad \text{,,} \times 19{\cdot}625 = 4750$$

$$N_{22} = \text{,,} \times 115 = 5{\cdot}35 \qquad T_{22} = \text{,,} \quad \text{,,} \times 21{\cdot}75 = 5260$$

$$N_{23} = \text{,,} \times 95{\cdot}7 = 4{\cdot}45 \qquad T_{23} = \text{,,} \quad \text{,,} \times 23{\cdot}875 = 5760$$

$$N_{24} = \text{,,} \times 79{\cdot}3 = 3{\cdot}68 \qquad T_{24} = \text{,,} \quad \text{,,} \times 26 = 6290$$

Diagram V. shows our standard power-curve (*bc*) and the characteristic for the 13-in. Ryder lathe with all-gear headstock. It is at once noticeable that nearly twice as much power is put into this headstock as is called for by our standard cut taken at the proper speed for high-speed steel. This is accounted for by the fact that, with an all-gear head in use, the belt can be run at a very high speed. (In this instance a 4½-in. belt is run at 1900 ft. per minute. With an effective tension of 75 lbs. per inch, this is equal to 18 G.H.P., whereas our standard G.H.P. for this size is 11 horse-power.) We have therefore drawn the characteristic of this lathe in relation to its own belt horse-power instead of to our standard curve.



Torque Speed Diagram for 13-In. Lathe (all-gear head).

DIAGRAM V.

The torques and spindle revolutions are tabulated below; the former having been deduced from data supplied by the makers with the allowances for friction, belt tension, etc., indicated in the schedule.

18-IN. ALL GEAR HEAD.

Pulley 18 *in. diameter* ; *belt* 4¼ *in. wide.*
Countershaft, 400 and 480 revolutions per minute.
Effective tension = 75 lbs. per inch width.

$$\text{Available torque at pulley} = \frac{75 \times 4\tfrac{1}{4} \times 18}{24} = 240.$$

$$N_1 = 480 \times \frac{26}{54} \times \frac{59}{68} = 200 \qquad T_1 = 240 \times \frac{480}{200} \times (0{\cdot}95)^2 = 518$$

$$N_2 = 400 \times \text{,,} = 168 \qquad T_2 = \text{,,} \times \frac{400}{168} \times \text{,,} = 518$$

$$N_3 = 480 \times \frac{20}{60} \times \frac{59}{68} = 139 \qquad T_3 = \text{,,} \times \frac{480}{139} \times \text{,,} = 746$$

$$N_4 = 400 \times \text{,,} = 116 \qquad T_4 = \text{,,} \times \frac{400}{116} \times \text{,,} = 746$$

$$N_5 = 480 \times \frac{26}{54} \times \frac{39}{88} = 102{\cdot}5 \qquad T_5 = \text{,,} \times \frac{480}{102{\cdot}5} \times \text{,,} = 1{,}010$$

$$N_6 = 400 \times \text{,,} = 85{\cdot}5 \qquad T_6 = \text{,,} \times \frac{400}{85{\cdot}5} \times \text{,,} = 1{,}010$$

$$N_7 = 480 \times \frac{20}{60} \times \frac{39}{88} = 71 \qquad T_7 = \text{,,} \times \frac{480}{71} \times \text{,,} = 1{,}460$$

$$N_8 = 400 \times \text{,,} = 59 \qquad T_8 = \text{,,} \times \frac{400}{59} \times \text{,,} = 1{,}460$$

$$N_9 = 480 \times \frac{26}{54} \times \frac{23}{104} = 51{\cdot}2 \qquad T_9 = \text{,,} \times \frac{480}{51{\cdot}2} \times \text{,,} = 2{,}025$$

$$N_{10} = 400 \times \text{,,} = 42{\cdot}6 \qquad T_{10} = \text{,,} \times \frac{400}{42{\cdot}6} \times \text{,,} = 2{,}025$$

$$N_{11} = 480 \times \frac{20}{60} \times \frac{23}{104} = 35{\cdot}3 \qquad T_{11} = \text{,,} \times \frac{480}{35{\cdot}3} \times \text{,,} = 2{,}940$$

$$N_{12} = 400 \times \text{,,} = 29{\cdot}4 \qquad T_{12} = \text{,,} \times \frac{400}{29{\cdot}4} \times \text{,,} = 2{,}940$$

$$N_{13} = 480 \times \frac{25}{54} \times \frac{59}{68} \times \left(\frac{26}{62} \times \frac{15}{51}\right) = 24{\cdot}9 \qquad T_{13} = 240 \times \frac{480}{24{\cdot}9} \times (0{\cdot}95)^4 = 3{,}750$$

$$N_{14} = 400 \times \text{,,} \times \text{,,} = 20{\cdot}7 \qquad T_{14} = \text{,,} \times \frac{400}{20{\cdot}7} \times \text{,,} = 3{,}750$$

$$N_{15} = 480 \times \frac{20}{60} \times \frac{59}{68} \times \text{,,} = 17{\cdot}2 \qquad T_{15} = \text{,,} \times \frac{480}{17{\cdot}2} \times \text{,,} = 5{,}430$$

$$N_{16} = 400 \times \text{,,} \times \text{,,} = 14{\cdot}3 \qquad T_{16} = \text{,,} \times \frac{400}{14{\cdot}3} \times \text{,,} = 5{,}430$$

$$N_{17} = 480 \times \frac{26}{54} \times \frac{39}{88} \times \text{,,} = 12{\cdot}68 \qquad T_{17} = \text{,,} \times \frac{480}{12{\cdot}68} \times \text{,,} = 7{,}370$$

$$N_{18} = 400 \times \text{,,} \times \text{,,} = 10{\cdot}55 \qquad T_{18} = \text{,,} \times \frac{400}{10{\cdot}55} \times \text{,,} = 7{,}370$$

$$N_{19} = 480 \times \frac{20}{60} \times \frac{39}{88} \times \text{,,} = 8{\cdot}75 \qquad T_{19} = \text{,,} \times \frac{480}{8{\cdot}75} \times \text{,,} = 10{,}650$$

$$N_{20} = 400 \times \text{,,} \times \text{,,} = 7{\cdot}27 \qquad T_{20} = \text{,,} \times \frac{400}{7{\cdot}27} \times \text{,,} = 10{,}650$$

$$N_{21} = 480 \times \frac{26}{54} \times \frac{23}{104} \times \text{,,} = 6{\cdot}3 \qquad T_{21} = \text{,,} \times \frac{480}{6{\cdot}3} \times \text{,,} = 14{,}800$$

$$N_{22} = 400 \times \text{,,} \times \text{,,} = 5{\cdot}25 \qquad T_{22} = \text{,,} \times \frac{400}{5{\cdot}25} \times \text{,,} = 14{,}800$$

$$N_{23} = 480 \times \frac{20}{60} \times \frac{23}{104} \times \text{,,} = 4{\cdot}35 \qquad T_{23} = \text{,,} \times \frac{480}{4{\cdot}35} \times \text{,,} = 21{,}450$$

$$N_{24} = 400 \times \text{,,} \times \text{,,} = 3{\cdot}62 \qquad T_{24} = \text{,,} \times \frac{400}{3{\cdot}62} \times \text{,,} = 21{,}450$$

Neglecting the second motion countershaft speed, the ratio of the characteristic area to that of our standard diagram is $\zeta = 2{\cdot}06$, which means that twice as much material can be machined off by this machine as could be by

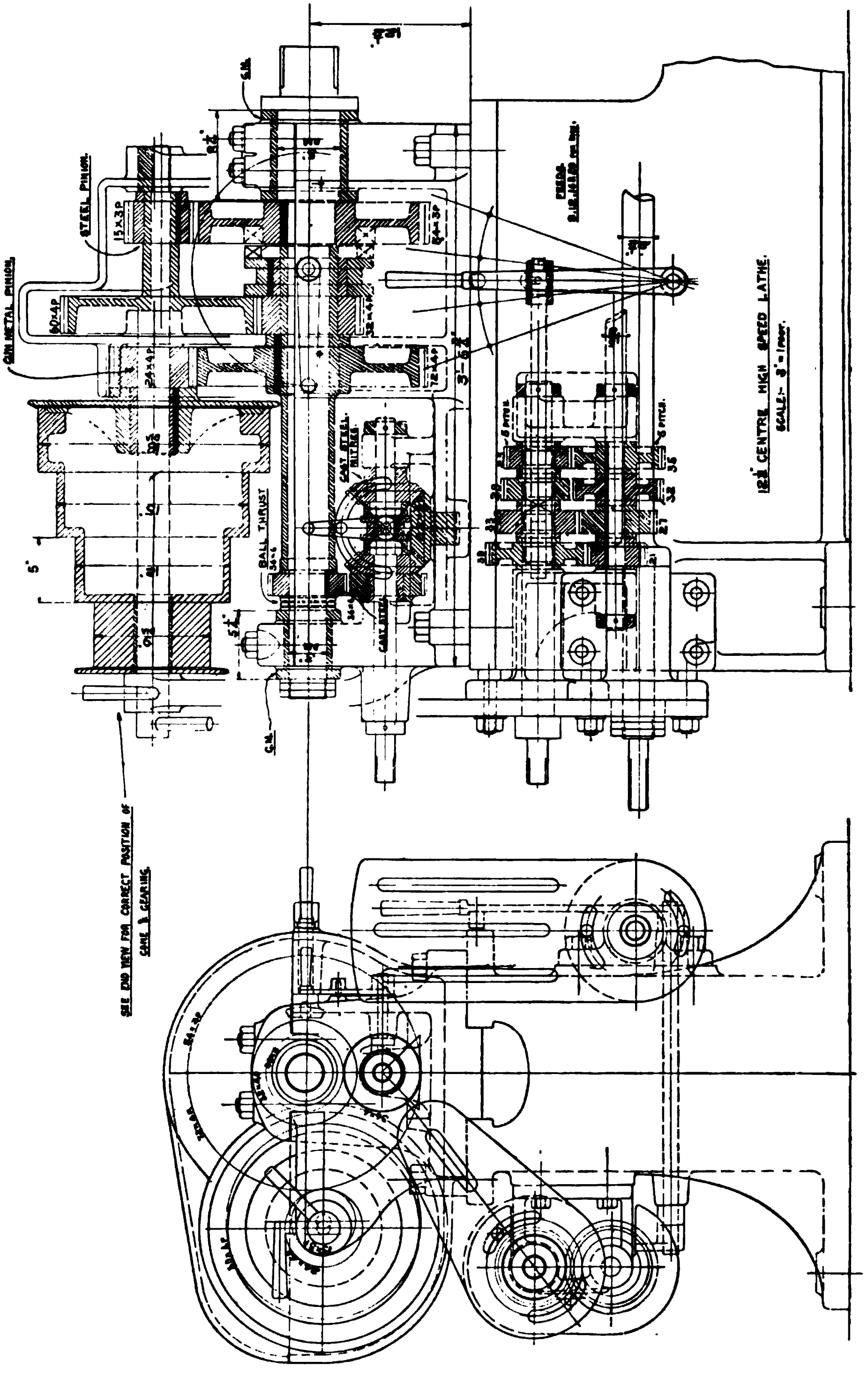


FIG. 249.—HEADSTOCK FOR 12½-IN CENTRES HIGH-SPEED LATHE.

a perfect lathe supplied only with our standard amount of 11 G.H.P. (*vide* Table XXV. p. 33). A saving of 7*s.* per ton of steel machined should be effected from this cause, provided the first cost of the lathe is about what our formula assumes it to be.

The second counter-motion speed is, in our opinion, of no use whatever, and, in fact, detracts from the weight-removing efficiency of the headstock. Its only value lies in the possibility of getting the slowest spindle speeds, thereby provided for use with carbon steel or with gapped work.

Examples VI. and VII.—**12½-in. and 18-in. Centres Lathes by the Miley Machine Tool Co., Ltd.** *Cone on side shaft,* 12½-*in. Lathe* (Fig. 249 and Diag. V.); 18-*in. Lathe* (Fig. 250 and Diag. VI.).

An improved design of headstock for high speed lathes by the Miley Machine Tool Co., Ltd., is shown in Fig. 249. The headstock reproduced is for a 12½-in. centre lathe, and the design is representative of that adopted by the above firm in lathes between 10½ and 18-in. centres.

The cone is here mounted on a side shaft, and motion is communicated to the spindle through a single or treble train of gear. The first motion pinion and cone are combined, and rotate freely on the concentric portion of the stationary shaft. The shaft is turned eccentrically between the two front bearings, and thereon is mounted the second driven wheel and third driving pinion; the latter being keyed to the boss of the former. The arrangement is such that by turning the shaft through a fraction of a revolution the last-mentioned wheel and pinion are withdrawn from mesh with those on the spindle; the first train (or single gear) only remaining in gear. The first-motion wheel and second driving pinion are keyed to a sleeve freely mounted upon the spindle. On the boss of the last pinion is keyed a positive clutch which engages with the front driving wheel keyed to the spindle. Thus, when running single gear, the clutch is engaged with the front driving wheel, and the wheels on the eccentric shaft are disengaged, and conversely when in treble gear. The gear ratios and spindle speeds, etc., for the 12½-in. headstock are given below, and the accompanying tables give the principal dimensions of the 10½, 12½, 15, and 18-in. lathes. The spindles in these machines are of mild steel, ground in the necks, which run in parallel gun-metal steps. A Hoffmann Ball Bearing is fitted to inside of the tail cheek to take the backward thrust on the spindle. The reversing motion to the screw-cutting feed is obtained through machine-cut cast steel mitre wheels placed between the headstock cheeks, and actuated by the handle on the outside of the casing.

The feed change mechanism consists of four trains of gear, either of which can be engaged by a sliding cotter controlled by the handle at the front of the lathe, as shown in Fig. 249.

12½-in. Centre High-Speed Lathe Head.
Countershaft to run 400 and 500 revolutions per minute.
Cone Pulley 9 in., 12 in., 15 in., and 18 in. diameter by 5 in. wide.
1st Gear $\frac{72}{24}$ = 3 to 1. 2nd Gear $\frac{72}{24}$ 4P. $\frac{66}{33}$ 4P. $\frac{40}{12}$ 3P. Total 20 to 1.

Revs. of cone	200	250	320	400	500	625	800	1000
Revs. of spindle in first gear .	66	83	106	133	166	208	266	333
Dia. turned at 70 ft. per min.	4 in.	3·2 in.	2·5 in.	2 in.	1·6 in.	1·2 in.	1 in.	0·8 in.
Revs. of spindle in second gear	10	12·5	16	20	25	31	40	50
Dia. turned at 70 ft. per min.	26 in.	20 in.	16 in.	13·3 in.	10 in.	8·6 in.	6·6 in.	5·3 in.

Principal Dimensions of the Miley Lathes for High-Speed Steels.

Height of centres	**10½** in.	**12½** in.	**15** in.	**18** in.
Length of bed	16 ft.	18 ft.	18 ft.	20 ft.
Width of bed across vees	17 in.	21 in.	24 in.	28 in.
Depth of bed	12 in.	14 in.	15 in.	15 in.
Admits between centres	10 ft. 6 in.	11 ft. 0 in.	10 ft. 3 in.	12 ft. 0 in.
Gap admits (diameter)	48 in.	52 in.	60 in.	68 in.
Gap admits (width)	16 in.	16 in.	18 in.	18 in.
Driving cone, number of speeds	4	4	4	4
Driving cone, largest and smallest diameters	6 in. & 14½ in.	9 in. & 18 in.	10 in. & 20 in.	14 in. & 26 in.
Driving cone, width of belt	4 in.	5 in.	5½ in.	6 in.
Fast headstock, front neck, diameter and length	3½ in. × 6 in.	5 in. × 8½ in.	5½ in. × 8½ in.	6 in. × 9 in.
Fast headstock, back neck, diameter and length	2½ in. × 4¾ in.	3 in. × 5½ in.	3½ in. × 5½ in.	4 in. × 7 in.
Ratio of gearing	3·2 to 1 29·8 to 1	3 to 1 20 to 1	3 to 1 21 to 1	4 to 1 22·5 to 1
Guide screw, diameter of	2¼ in.	2½ in.	2¾ in.	3 in.
Loose head and spindle, diameter of	2½ in.	3½ in.	4 in.	4 in.
Speeds of countershaft, revolutions per minute	400–500	400–500	400–500	400–500
Sizes of face-plates	20–36 in.	24–48 in.	30–54 in.	36–60 in.

The headstock of the 18-in. centre high-speed lathe by the same firm is shown in Fig. 250 (rear view) at the end of the book.

The driving wheel, instead of being mounted on the spindle as in the 12½-in. lathe, is fixed to the back of the face-plate, no power whatever being transmitted through the spindle. The gear throughout is machine-cut out of the solid. The spindle is of steel, having a diameter of 8 inches at the front and 4 inches at the rear neck. The thrust is taken on a ball bearing fitted at the rear of the spindle. Particulars of spindle and countershaft speeds, etc., are given in the table below. The face-plate is fitted with removable jaws so that it can be used for chuck or face-plate work. The feeds to the saddle and rests are $\frac{1}{32}$ in., $\frac{1}{16}$ in., $\frac{1}{8}$ in., and $\frac{1}{4}$ in. per revolution of spindle. The loose headstock has a steel spindle 4 inches diameter, of equal diameter throughout, and being threaded at one end, is adjusted by means of a rotary nut and hand wheel. The screwed end projects through the barrel end, thus ensuring a rigid support to the spindle when in its extreme outward position.

18-in. Centre High-Speed Lathe Head.

Countershaft to run 300 and 500 revolutions per minute.

Cone Pulley 14 in., 18 in., 22 in., and 26 in. diameter by 6 in. wide.

1st Gear $\frac{68}{11}$ × 1⅜ in. = 6·18 to 1. 2nd Gear $\frac{68}{11}$ × 1⅜ in. [illegible] × 2¼. [illegible] × 2½ P. Total 44·7 to 1.

Revs. of cone	161·5	245·4	269	366·6	409	557	611	928·5
Revs. of spindle in first gear	26	39·7	43·5	59·3	66	90	98	150
Dia. turned at 70 ft. per min.	10·2 in.	7 in.	6·2 in.	4·6 in.	4·1 in.	3 in.	2·7 in.	1·8 in.
Revs. of spindle in second gear	3·6	5·4	6	8	9	12·4	13·6	20·7
Dia. turned at 70 ft. per min.	74 in.	50·7 in.	44·6 in.	33·2 in.	29·2 in.	21·2 in.	19·7 in.	13·2 in.

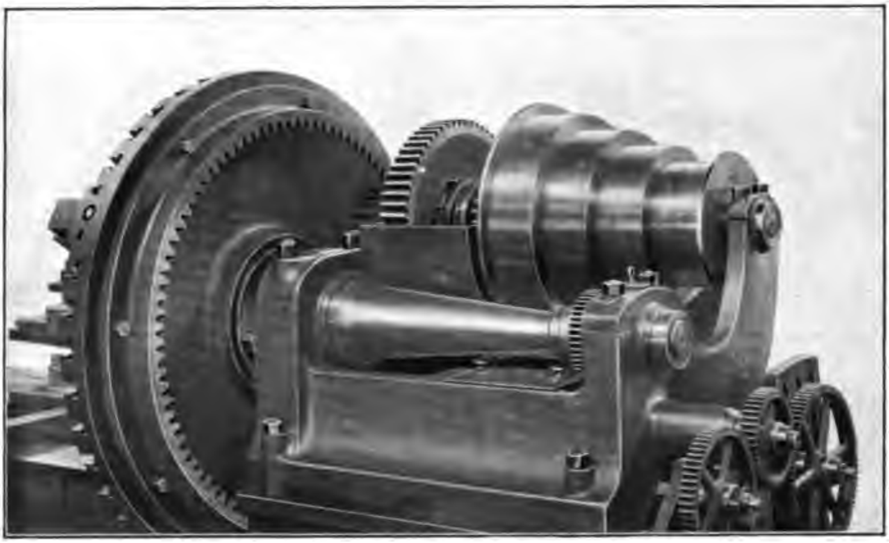


Fig. 250.—Headstock for 18-in. Centres High-speed Lathe (Miley).

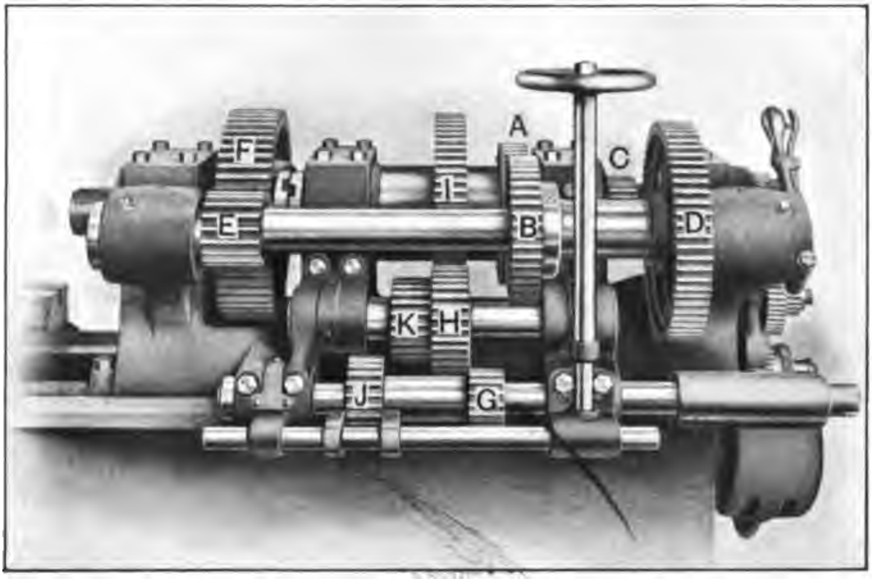
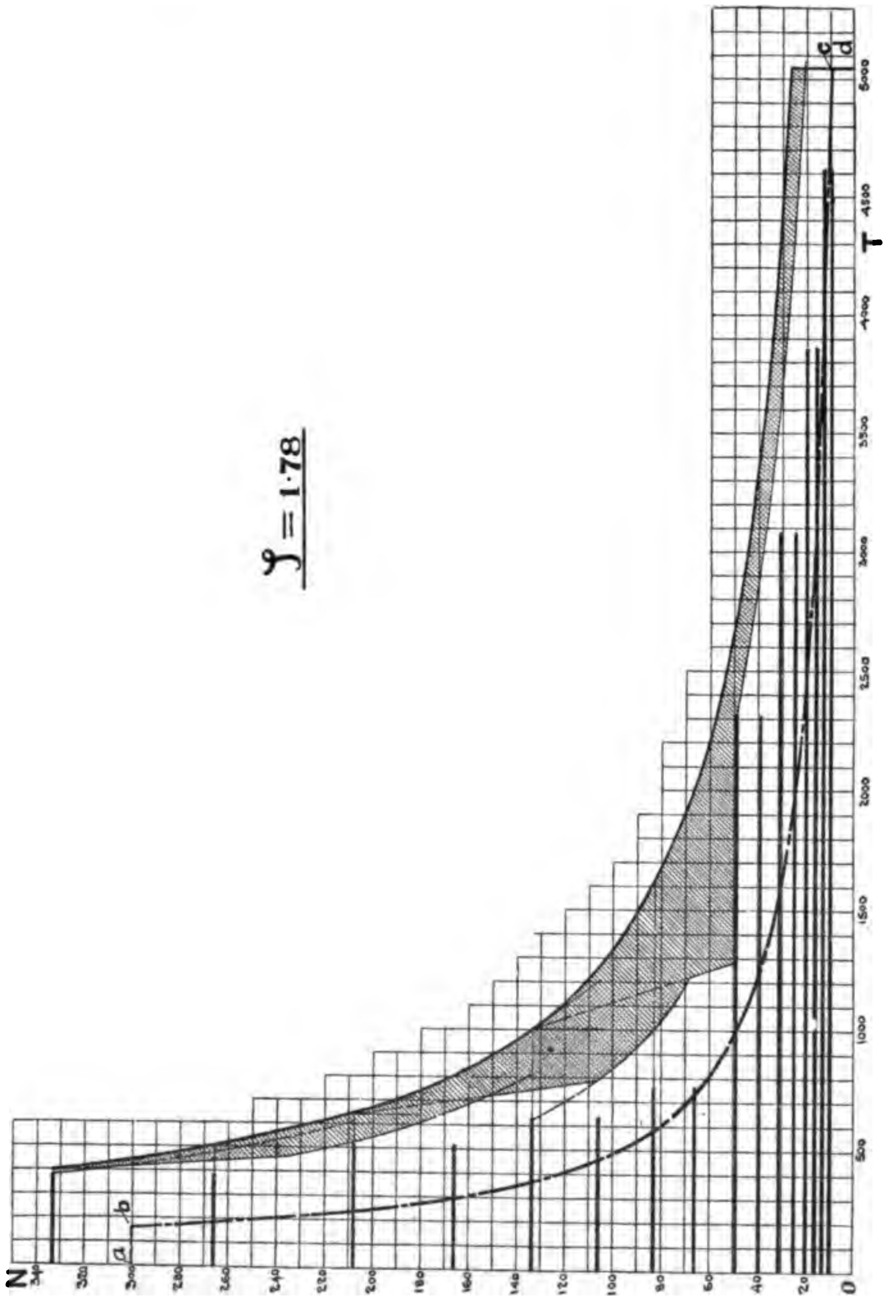


Fig. 254.—All-gear Headstock for 10-in. Centres High-speed Lathe (Lodge and Shipley).

Diagrams VI. and VII. give the lathe characteristics for these two lathes, whilst the schedules annexed give the torques corrected for friction, as in the previous examples.



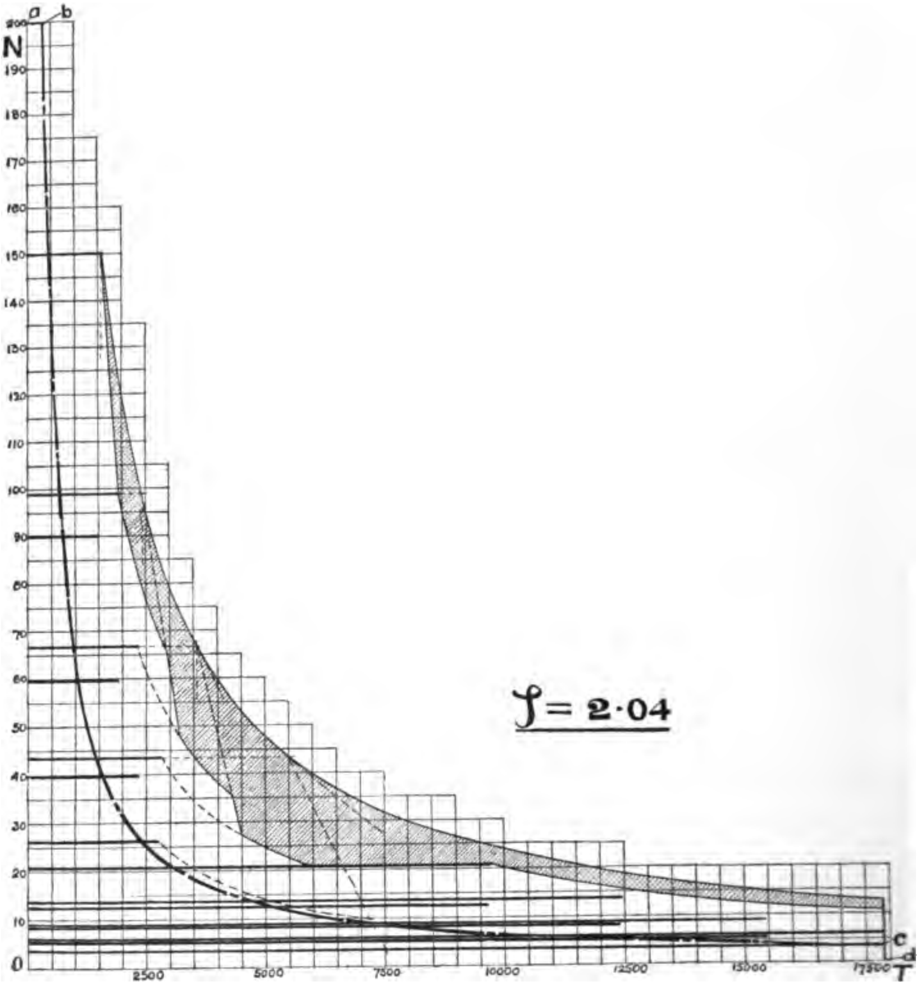
Torque Speed Diagram for 12-in. Lathe (cone off spindle).

DIAGRAM VI.

The power is much greater than what our standard calls for. A convenient way of finding their weight-removing efficiencies (ζ) is to compare the actual torques provided at the tool with those of a perfect lathe of the maximum belt power supplied on any speed (one, that is, possessing every spindle speed within the range of the machine, and the same effective power at the tool at every such speed), and to draw the characteristic relatively to that. The ratio of the unshaded area to our standard area *oabcd* then gives ζ the required efficiency.

For the 12½-in. lathe $\zeta = 1{\cdot}78$, and for the 18-in. $\zeta = 2{\cdot}04$. These efficiencies correspond to savings of about 5*s.* and 7*s.* per ton machined respectively.

The lower countershaft speeds (except for the lowest spindle speeds) are again shown to be of no value.



Torque Speed Diagram for 18-in. Lathe (cone off spindle).

DIAGRAM VII.

12½-IN. LATHE. CONE OFF SPINDLE.

Cone step diameters, 9, 12, 15, 18 × 5 ins. wide.
Countershaft speeds, 400 and 500 revolutions per minute.
Belt width taken at 4¾ ins. Effective tension, 75 lbs. per inch width.
Gear ratios—3 single and 20·25 treble to 1.

$$N_1 = 500 \times \frac{18}{9} \times \frac{1}{3} = 333 \qquad T_1 = \left(\frac{4{\cdot}75 \times 75 \times 3}{24}\right) \times 9 \times 0{\cdot}95 = 380$$

$$N_2 = 400 \times \frac{18}{9} \times \text{,,} = 266 \qquad T_2 = \text{,,} \times 9 \times \text{,,} = 380$$

$$N_3 = 500 \times \frac{15}{12} \times \text{,,} = 208 \qquad T_3 = \text{,,} \times 12 \times \text{,,} = 508$$

$$N_4 = 400 \times \frac{15}{12} \times \text{,,} = 166 \qquad T_4 = \text{,,} \times 12 \times \text{,,} = 508$$

$$N_5 = 500 \times \frac{12}{15} \times \text{,,} = 133 \qquad T_5 = \text{,,} \times 15 \times \text{,,} = 635$$

$$N_6 = 400 \times \frac{12}{15} \times \text{,,} = 106 \qquad T_6 = \text{,,} \times 15 \times \text{,,} = 635$$

$$N_7 = 500 \times \frac{9}{18} \times \text{,,} = 83 \qquad T_7 = \text{,,} \times 18 \times \text{,,} = 760$$

$$N_8 = 400 \times \frac{9}{18} \times \text{,,} = 66 \qquad T_8 = \text{,,} \times 18 \times \text{,,} = 760$$

$$N_9 = 500 \times \frac{18}{9} \times \frac{1}{20{\cdot}25} = 49{\cdot}4 \qquad T_9 = \left(\frac{(4{\cdot}75 \times 75 \times 20{\cdot}25)}{24}\right) \times 9 \times (0{\cdot}95)^2 = 2310$$

$$N_{10} = 400 \times \frac{18}{9} \times \text{,,} = 39{\cdot}6 \qquad T_{10} = \text{,,} \times 9 \times \text{,,} = 2310$$

$$N_{11} = 500 \times \frac{15}{12} \times \text{,,} = 30{\cdot}8 \qquad T_{11} = \text{,,} \times 12 \times \text{,,} = 3080$$

$$N_{12} = 400 \times \frac{15}{12} \times \text{,,} = 24{\cdot}7 \qquad T_{12} = \text{,,} \times 12 \times \text{,,} = 3080$$

$$N_{13} = 500 \times \frac{12}{15} \times \text{,,} = 19{\cdot}75 \qquad T_{13} = \text{,,} \times 15 \times \text{,,} = 3860$$

$$N_{14} = 400 \times \frac{12}{15} \times \text{,,} = 15{\cdot}7 \qquad T_{14} = \text{,,} \times 15 \times \text{,,} = 3860$$

$$N_{15} = 500 \times \frac{9}{18} \times \text{,,} = 12{\cdot}3 \qquad T_{15} = \text{,,} \times 18 \times \text{,,} = 4560$$

$$N_{16} = 400 \times \frac{9}{18} \times \text{,,} = 9{\cdot}78 \qquad T_{16} = \text{,,} \times 18 \times \text{,,} = 4560$$

18-IN. LATHE. CONE OFF SPINDLE.

Cone step diameters, 14, 18, 22, 26 × 6 ins. wide.
Countershaft speeds, 300 and 500 revolutions per minute.
Belt width taken, 5¾ ins. Effective tension = 75 lbs. per inch width.
Gear ratios—6·18 single and 44·7 treble to 1.

$$N_1 = 500 \times \frac{26}{14} \times \frac{1}{6{\cdot}18} = 150 \qquad T_1 = \left(\frac{5{\cdot}75 \times 75}{24}\right) \times 6{\cdot}18 \times 0{\cdot}95 \times 14 = 1480$$

$$N_2 = \text{,,} \times \frac{22}{18} \times \text{,,} = 99 \qquad T_2 = \text{,,} \quad \text{,,} \quad \text{,,} \times 18 = 1900$$

$$N_3 = \text{,,} \times \frac{18}{22} \times \text{,,} = 66{\cdot}3 \qquad T_3 = \text{,,} \quad \text{,,} \quad \text{,,} \times 22 = 2330$$

$$N_4 = \text{,,} \times \frac{14}{26} \times \text{,,} = 43{\cdot}6 \qquad T_4 = \text{,,} \quad \text{,,} \quad \text{,,} \times 26 = 2750$$

$$N_5 = 300 \times \frac{26}{14} \times \text{,,} = 90 \qquad T_5 = \text{,,} \quad \text{,,} \quad \text{,,} \times 14 = 1480$$

$$N_6 = \text{,,} \times \frac{22}{18} \times \text{,,} = 59{\cdot}4 \qquad T_6 = \text{,,} \quad \text{,,} \quad \text{,,} \times 18 = 1900$$

$$N_7 = 300 \times \frac{18}{22} \times \frac{1}{6{\cdot}18} = 39{\cdot}8 \qquad T_7 = \left(\frac{5{\cdot}75 \times 75}{24}\right) \times 6{\cdot}18 \times 0{\cdot}95 \times 22 = 2330$$

$$N_8 = \text{,,} \times \frac{14}{26} \times \text{,,} = 26{\cdot}2 \qquad T_8 = \text{,,} \quad \text{,,} \quad \text{,,} \times 26 = 2750$$

$$N_9 = 500 \times \frac{26}{14} \times \frac{1}{44{\cdot}7} = 20{\cdot}75 \qquad T_9 = \text{,,} \quad \times 44{\cdot}7 \times (0{\cdot}95)^3 \times 14 = 9650$$

$$N_{10} = \text{,,} \times \frac{22}{18} \times \text{,,} = 13{\cdot}7 \qquad T_{10} = \text{,,} \quad \text{,,} \quad \text{,,} \times 18 = 12400$$

$$N_{11} = \text{,,} \times \frac{18}{22} \times \text{,,} = 9{\cdot}15 \qquad T_{11} = \text{,,} \quad \text{,,} \quad \text{,,} \times 22 = 15400$$

$$N_{12} = \text{,,} \times \frac{14}{26} \times \text{,,} = 6{\cdot}08 \qquad T_{12} = \text{,,} \quad \text{,,} \quad \text{,,} \times 26 = 17950$$

$$N_{13} = 300 \times \frac{26}{14} \times \text{,,} = 12{\cdot}43 \qquad T_{13} = \text{,,} \quad \text{,,} \quad \text{,,} \times 14 = 9650$$

$$N_{14} = \text{,,} \times \frac{22}{18} \times \text{,,} = 8{\cdot}16 \qquad T_{14} = \text{,,} \quad \text{,,} \quad \text{,,} \times 18 = 12400$$

$$N_{15} = \text{,,} \times \frac{18}{22} \times \text{,,} = 5{\cdot}5 \qquad T_{15} = \text{,,} \quad \text{,,} \quad \text{,,} \times 22 = 15400$$

$$N_{16} = \text{,,} \times \frac{14}{26} \times \text{,,} = 3{\cdot}62 \qquad T_{16} = \text{,,} \quad \text{,,} \quad \text{,,} \times 26 = 17950$$

EXAMPLE 8.—**32-in. Lathe for High-speed Steel by Hulse & Co.** Motor driven, all-gear head (Fig. 251 and Diagram VIII.).

This 32-in. centre patent, sliding, surfacing, screw-cutting, and taper-turning lathe is designed to admit 58 ft. between the centres and to swing 46 ins. diameter over the carriages.

The fast headstock, depicted by Fig. 251 (for general arrangement see Fig. 170 at end of book), is furnished with a 50 H.P. motor having a speed variation from 300 to 900 revolutions per minute, and mounted on a projection from the bed at the front of the headstock. A raw hide pinion (37 T × $1\frac{1}{8}$ in. *p*) keyed to the armature shaft meshes with the large spur wheel (100 T × $1\frac{1}{8}$ in. *p*) on the first-motion shaft. Keyed to this shaft is a pinion having 30 teeth by $1\frac{1}{4}$ in. pitch, which gears with the spur wheel on the sleeve carried by the face-plate pinion shaft. This sleeve is bushed and free to rotate independently of the shaft, whilst it has keyed to it the three spur wheels and locking plate. (The sleeve is identical with the cone sleeve of the ordinary lathe with cone on side shaft.) The two pinions at the tail end of sleeve mesh with two spur wheels on the third-motion shaft, which also carries a pinion and gears with the front driving wheel. Neglecting the pair of wheels between the motor and first motion shaft, there is one double gear—

$$\frac{75}{30} \times \frac{102}{21}$$

and two quadruple gears—

$$\text{1st} \quad \frac{75}{30} \times \frac{50}{52} \times \frac{64}{21} \times \frac{102}{21},$$

$$\text{2nd} \quad \frac{75}{30} \times \frac{75}{27} \times \frac{64}{21} \times \frac{102}{21},$$

giving a gear ratio between the motor and face-plate of 32·8, 96·1, and 277 to 1, and corresponding revolutions to the spindle of 27·3 to 1·07.

It will be noticed that the pinions have a large number of teeth, none being

less than 21. The gear throughout is machine-cut, and all the pinions are of mild steel and case-hardened. All the bearings for revolving parts are bushed with brass. The main spindle runs in gun metal capped bearings, the rear bearing being fitted with collars to take the backward thrust of the spindle, whilst the washer and nut within the cheeks and the nuts and washers at the tail of headstock take the pull on the spindle when cutting towards the loose headstock. Supplementary cheeks have been added to the back of the headstock to stiffen it for heavy cutting. The headstock is provided with square lips which bear on the vertical faces of the inside ways of the bed to resist cross-wind, and is held to it by eight short screws, two of which are placed immediately at the front of headstock and beneath the face-plate wheel. The face-plate chuck is 5 ft. 4 ins. diameter, fitted with four independent, hardened steel jaws, tongued and grooved into the plate, and actuated by four steel square-threaded screws. These screws are held at both ends in order that they may always be in tension.

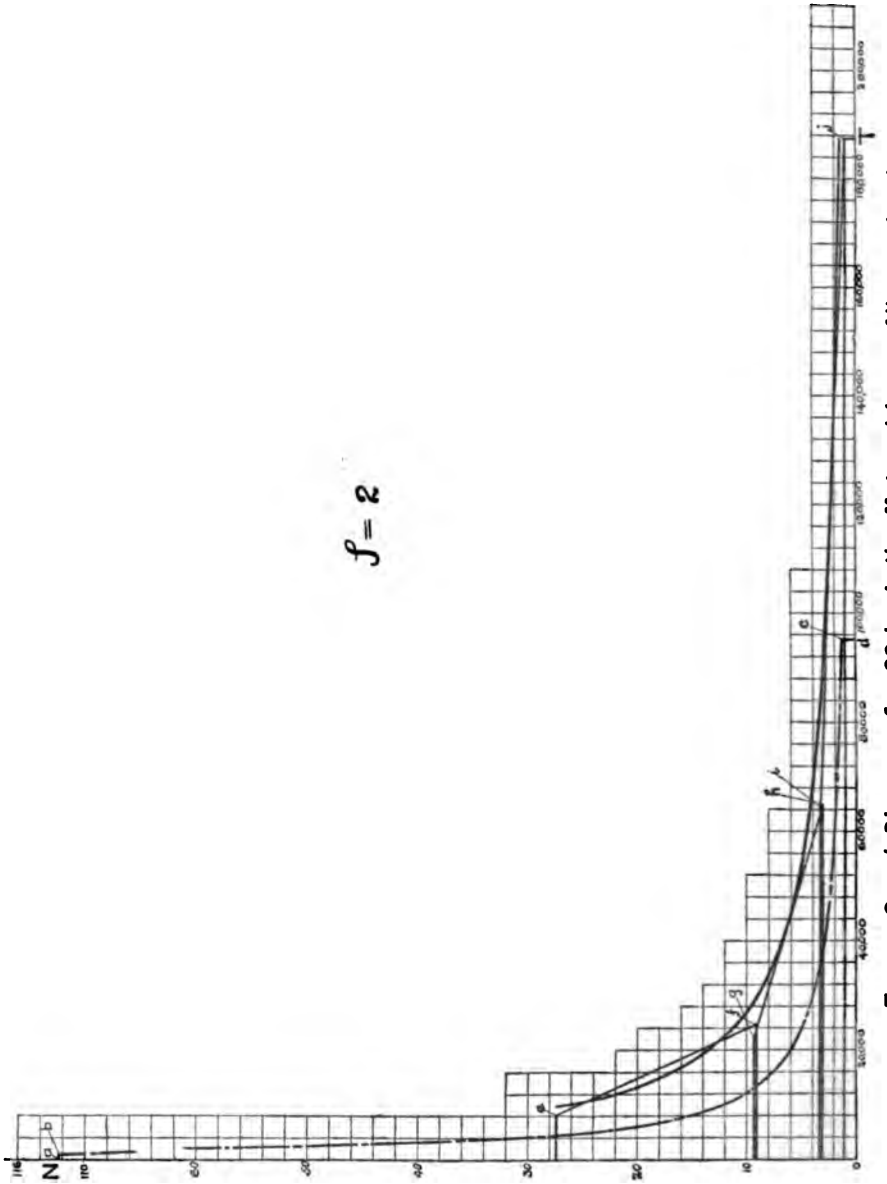
The bed is of the four-girder type with four ways, is 69 ft. 3 ins. long, 6 ft. 6 ins. wide, 21 ins. deep, in three lengths, and provided with two non-rotating steel guide screws, each in one length. Four independent carriages are mounted on the bed—two at the back and two at the front—each fitted with compound slide rests, rotating nut, and reversing mechanism. The front carriages are fitted with tapering mechanism, and the front rests with worm and wheel adjustment to the screw in swivel slide.

Particulars of Wheels for 32-inch Lathe Fast Headstock.

Number of Teeth.	Pitch.	Bottom Diameter of Teeth.	Top Diameter of Teeth.	Pitch Diameter.	Material.	Centres.
	ins.	ins.	ins.	ins.		ins.
37	$1\frac{1}{8}$	$12\frac{27}{64}$	$13\frac{31}{32}$	$13\frac{1}{4}$	Raw Hide	$24\frac{17}{32}$
100	,,	$34\frac{63}{64}$	$36\frac{17}{32}$	$35\frac{13}{16}$	C. I.	,,
30	$1\frac{1}{4}$	$11\frac{47}{64}$	$12\frac{47}{64}$	$11\frac{15}{16}$	M. S. Case Hardened.	$20\frac{29}{32}$
75	,,	$28\frac{15}{16}$	$30\frac{21}{32}$	$29\frac{55}{64}$	C. I.	,,
21	$1\frac{1}{2}$	$8\frac{59}{64}$	$10\frac{63}{64}$	$10\frac{1}{32}$	M. S. Case Hardened.	$20\frac{5}{16}$
64	,,	$29\frac{29}{64}$	$31\frac{33}{64}$	$30\frac{9}{16}$	C. I.	,,
27	$1\frac{1}{4}$	$9\frac{27}{32}$	$11\frac{35}{64}$	$10\frac{3}{4}$	M. S. Case Hardened.	$20\frac{5}{16}$
75	,,	$28\frac{39}{64}$	$30\frac{41}{64}$	$29\frac{27}{32}$	C. I.	,,
52	,,	$19\frac{49}{64}$	$21\frac{31}{64}$	$20\frac{11}{16}$	,,	$20\frac{5}{16}$
50	,,	$18\frac{31}{32}$	$20\frac{11}{16}$	$19\frac{57}{64}$	,,	,,
21	$1\frac{3}{4}$	$10\frac{27}{64}$	$12\frac{13}{16}$	$11\frac{45}{64}$	M. S. Case Hardened.	$34\frac{1}{4}$
102	,,	$55\frac{17}{32}$	$57\frac{59}{64}$	$56\frac{13}{16}$	M. S.	,,
15	4 per in.	$3\frac{3}{16}$	$4\frac{1}{4}$	$3\frac{3}{4}$	M. S.	,,

Diagram VIII. shows our standard power curve (*bc*) for this size of lathe, and the characteristic of the machine. As the headstock is driven by an electric motor of 50 B.H.P. against our standard of 24 B.H.P., the torque-speed curves, as given by the sloping straight lines *ef*, *qh*, and *ij*, are altogether external to our standard curve. The spindle speed has continuous variations between these limits.

The value of ζ is about 2; so that weight removal can take place at our standard rate upon each of two tools working together.



Torque Speed Diagram for 32-in. Lathe Motor driven. All-gear head

DIAGRAM VIII.

According to our Fig. 241, the economy to be expected by this great supply of power, and the accompanying possibility of continuously varying the spindle speed, is about nine or ten shillings per ton machined, upon the cost of cutting with a 32-inch lathe provided with only half the power but possessing every possible spindle speed within the range.

Our highest and lowest spindle speeds are 112·5 and 1·27; whilst Messrs. Hulse make the highest spindle speed 27·5 and the lowest 0·5 revolutions per minute.

The particulars of torque motor speeds, etc., are given on the annexed schedule.

32-*inch Centre Motor Driven Lathe. Gear Ratios: Treble* 32·8 *to* 1. *Quintuple* 96·2 *and* 278 *to* 1. *Motor Speeds* 300 *to* 900 *Revolutions per Minute. Motor torques corresponding to these speeds* 875 *and* 300 *lb.-feet respectively.*

$$N_e = 900 \times \frac{1}{32{\cdot}8} = 27{\cdot}4 \qquad T_e = 300 \times 32{\cdot}8 \times (0{\cdot}95)^3 = 8420$$

and every intermediate speed and torque

$$N_f \ldots = 300 \times \frac{1}{32{\cdot}8} = 9{\cdot}13 \qquad T_f = 875 \times 32{\cdot}8 \times (0{\cdot}95)^3 = 24{,}500$$

$$N_g \ldots = 900 \times \frac{1}{96{\cdot}2} = 9{\cdot}35 \qquad T_g = 300 \times 96{\cdot}2 \times (0{\cdot}95)^5 = 22{,}250$$

and every intermediate speed and torque

$$N_h \ldots = 300 \times \frac{1}{96{\cdot}2} = 3{\cdot}12 \qquad T_h = 875 \times 96{\cdot}2 \times (0{\cdot}95)^5 = 64{,}800$$

$$N_i \ldots = 900 \times \frac{1}{278} = 3{\cdot}21 \qquad T_i = 300 \times 278 \times (0{\cdot}95)^5 = 64{,}200$$

and every intermediate speed and torque

$$N_j = 300 \times \frac{1}{278} = 1{\cdot}07 \qquad T_j = 875 \times 278 \times (0{\cdot}95)^5 = 187{,}500$$

EXAMPLE No. IX.—**Lathe of 1500 mm. (60 in.) Centres by Ernst Schiess, Düsseldorf** (Fig. 252 and Diagram IX.).

This example is that of a 1500 mm. (60 in.) centre lathe by Ernst Schiess, Düsseldorf, intended for turning heavy crank shafts and forgings with high-speed steel. The fast headstock is 2700 mm. (106 in.) long and is driven by a 50 H.P. variable speed motor, built on a base-plate bolted to the bed, through belting and a series of disengaging gears, whereby twenty different speeds are obtained to the spindle. A friction clutch is provided so that the lathe can be stopped without having to stop the motor. The face-plate is 3000 mm. (10 ft.) diameter and is fitted with four adjustable hardened steel jaws. The main spindle runs in adjustable phosphor bronze bearings, the front diameter of which is 450 mm. (18 in.) and 600 mm. (24 in.) long, whilst the rear bearing is 300 mm. (12 in.) diameter, and 385 mm. (15 in.) long. The arrangement of the gear is shown in Fig. 252, and the speeds obtained are as follows :—

Motor revolutions per minute.	1st Double gear.	Spindle revolutions per minute.	2nd Double gear.	Spindle revolutions per minute.
825 275	$\frac{21}{90} \times \frac{25}{74} = \frac{1}{12{\cdot}7}$	65 50 38 29 22	$\frac{21}{90} \times \frac{17}{135} = \frac{1}{34}$	24 18·8 14 10·7 8
	1st Quadruple gear.		**2nd Quadruple gear.**	
825 275	$\frac{21}{90} \times \frac{25}{92} \times \frac{44}{48} \times \frac{17}{135} = \frac{1}{137}$	6 4·7 3·6 2·7 2	$\frac{21}{90} \times \frac{25}{92} \times \frac{17}{75} \times \frac{17}{135} = \frac{1}{553}$	1·5 1·2 0·9 0·7 0·5

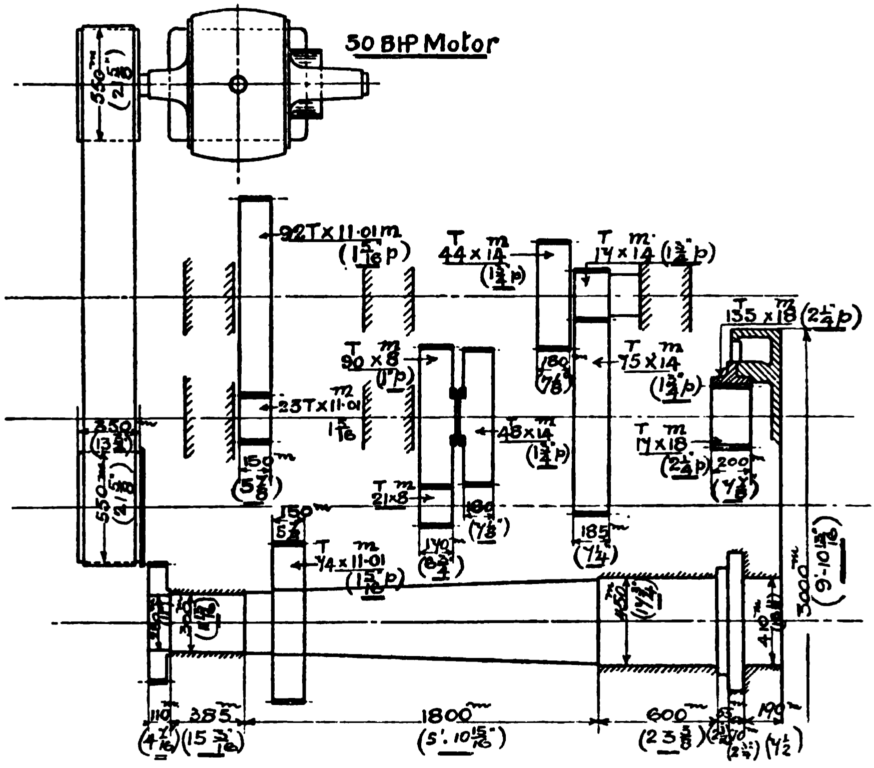
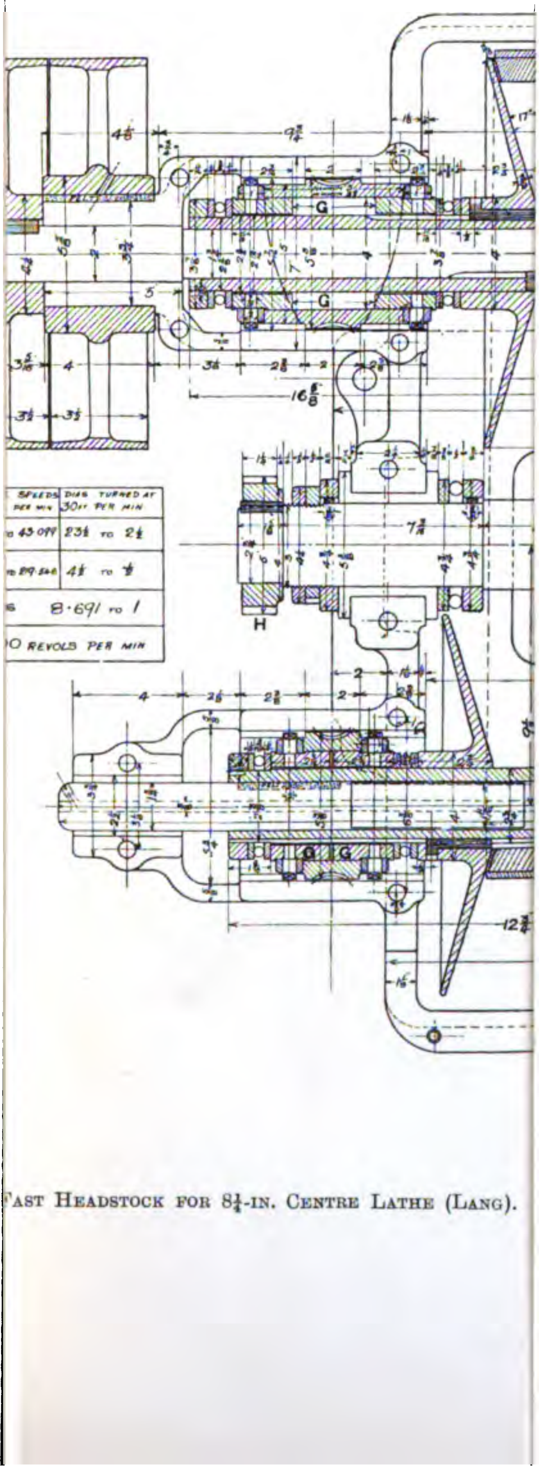


FIG. 252.—ARRANGEMENT OF HEADSTOCK GEAR IN 1500 MM. CENTRES LATHE.

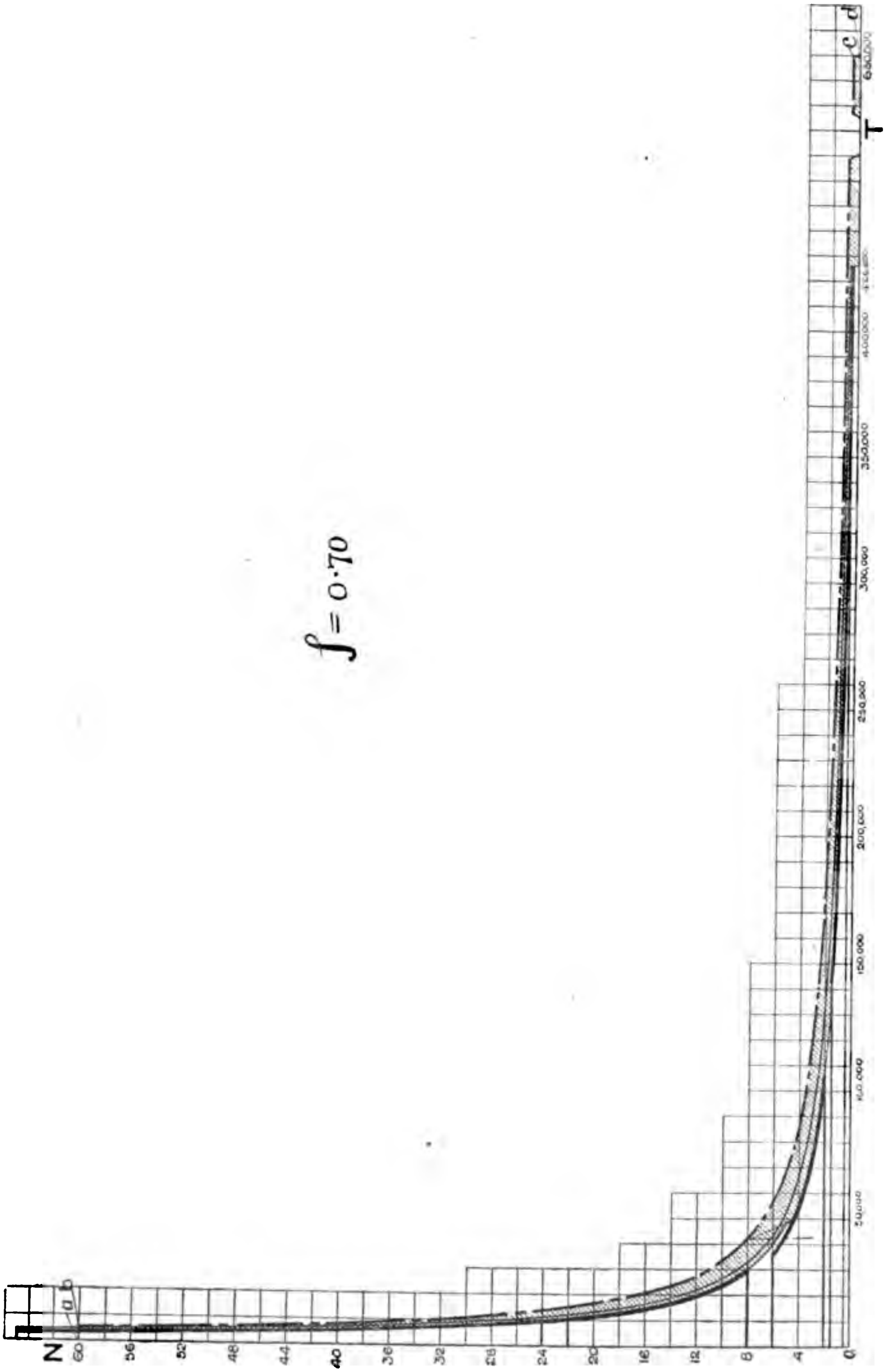
The headstock is mounted on a four way, four webbed bed 3100 mm. (122 in.) over ways, and 700 mm. (27½ in.) deep, whereon are fitted four carriages, two on the front ways and two on the back; each provided with compound slide rests. The slide can be quickly withdrawn. The lathe is of powerful construction throughout, and admits 10,000 mm. (33 ft.) between centres. All gear wheels have machine-cut teeth; the pinions, driving



ast Headstock for 8¼-in. Centre Lathe (Lang).

wheels, racks, and bevel wheels are of steel. All quick-running shafts are provided with adjustable phosphor bronze bearings.

The weight of this machine complete, including two steady and one follower rests, four face-plate jaws, etc., is approximately 126 tons.



Torque Speed Diagram for 60-in. Motor-belt driven Lathe.

DIAGRAM IX.

60-inch Centre Motor Driven Lathe. Gear Ratios: 12·7 *and* 34 *to* 1. *Quadruple* 137 *and* 553 *to* 1. *Constant horse-power variable speed motor from* 275 *to* 825 *revolutions per minute.*

$$N_1 = 825 \times \frac{1}{2{\cdot}7} = 66 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 66} \times (0{\cdot}95)^2 = 3640$$

and every intermediate speed and torque

$$N = 275 \times \frac{1}{12{\cdot}7} = 22 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 22} \times (0{\cdot}95)^2 = 10{,}750$$

$$N.. = 825 \times \frac{1}{34} = 24 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 24} \times (0{\cdot}95)^2 = 9850$$

and every intermediate speed and torque

$$N.. = 275 \times \frac{1}{34} = 8 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 8} \times (0{\cdot}95)^2 = 29{,}600$$

$$N = 825 \times \frac{1}{137} = 6 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 6} \times (0{\cdot}95)^4 = 35{,}500$$

and every intermediate speed and torque

$$N.. = 275 \times \frac{1}{137} = 2 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 2} \times (0{\cdot}95)^4 = 106{,}500$$

$$N.. = 825 \times \frac{1}{553} = 1{\cdot}5 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 1{\cdot}5} \times (0{\cdot}95)^4 = 143{,}000$$

and every intermediate speed and torque

$$N_2 = 275 \times \frac{1}{553} = 0{\cdot}5 \qquad T = \frac{50 \times 33{,}000}{2\pi \times 0{\cdot}5} \times (0{\cdot}95)^4 = 426{,}000$$

The annexed schedule shows the torques, speeds, etc., and Diagram IX. gives the characteristic for the Schiess lathe, our standard power curve *bc* for a 60-in. lathe being also drawn. The value of ζ is 0·70, the motor being of only 50 B.H.P., instead of the 74 B.H.P. which we should specify. This efficiency corresponds, according to our Fig. 241, with an extra expenditure incurred of about 4*s.* or 5*s.* per ton machined as compared with the cost (about 19*s.* per ton) for cutting with a perfect lathe of 74 B.H.P.

If a lathe of this size machines off 1000 tons of steel shavings in one year (and it may be expected to do much more), the above extra expenditure amounts to £250 per annum; and this sum is needlessly expended because of the lathe being underpowered. Similar losses have been shown to be incurred in some of the other lathes illustrated, owing to various defects in their design; *e.g.* improper gaps in the spindle speeds and undue diminution of the net pulling power in the middle ranges of spindle speed.

These examples have shown that the use of the torque-revolution characteristic, together with the standard-power curve of a lathe, provides a convenient means of studying any proposed design and eliminating its faults.

APPENDIX

LATHES BY GREENWOOD AND BATLEY.

ON Figs. 129 and 140, pp. 226 and 234, are shown details of the fast headstock and feed-change gear for an 8-in. centre lathe by Greenwood and Batley, Leeds. The driving cone has three steps—9½-in., 13¼-in., and 17-in. diameter × 3¾ in. wide, which with one set of double gear, having a ratio of 6 to 1, and two countershaft speeds, gives the spindle speeds shown in the accompanying table:—

Cone on countershaft, diameter used	9½ in.	9½ in.	13¼ in.	13¼ in.	17 in.	17 in.
Cone on spindle, diameter of speed used	17 in.	17 in.	13¼ in.	13¼ in.	9½ in.	9½ in.
Revolutions per minute of countershaft . . .	145	200	145	200	145	200

Revolutions per minute of spindle.

With gearing	13·5	18·6	24·6	33·3	43·2	60
Without gearing . . .	81	111·7	145	200	260	360

The saddle and slide-rests for this machine are shown in Fig. 154, and described on pp. 254 and 255, whilst the loose headstock is shown in Fig. 194, p. 306.

LATHE BY JOHN LANG AND SONS.

Fig. 253 shows the fast headstock for an 8¼-in. centre lathe by John Lang and Sons, Johnstone. The feed-change gear, the saddle and slide-rests, and the loose headstock for this machine have already been shown by Figs. 144 (p. 238), 155 (p. 256), and 195 (p. 314).

The fast headstock is arranged to be driven direct from the main shaft or motor, no counter-motion being necessary.

A fast and loose pulley is mounted on the first cone shaft at the rear of the headstock, and two expanding cones, keyed to this shaft, drive two similar cones on the second cone shaft situated at the front of the headstock. A pinion C is keyed to the right-hand half of the latter cone, but free to allow the cone to slide in a lateral direction when it is opening or closing, whilst the pinion is held in its position by a protruding collar which bears against the wheel D on the spindle. The cone and wheel F are free to rotate on the shaft, but capable of being locked to it by means of a double-faced clutch. The wheel D is keyed to the sleeve of the pinion E, and is independent of the spindle, the wheel B being keyed to it. The wheels are continually in gear, but the arrangement of the clutch is such that only one set can be positively engaged at one time. When running single gear the clutch on the second cone shaft is engaged with the pinion C, and the pinion A being keyed to the shaft drives the wheel B on the spindle, the wheels D, E, and F then running idle.

For the treble gear the clutch is engaged with the wheel F, and the drive is then through the wheels C, D, E, F, A, B. The cones are made to open and close simultaneously by means of cams cut on the bosses of the worm wheels. The worm wheels are actuated by a hand wheel keyed to the worm shaft as shown in end elevation. The cone has a long sleeve, one half being fixed to it, whilst the other half cone, although keyed to the sleeve, is free to slide along it. At the end of the cone and sleeve is a ball-thrust bearing which bears against the sliding sleeves G, carrying the worm wheels and thrust bowls. As the worm wheels are turned round, the cam, in the one case, thrusts the bowls and sleeves G outwards; the sleeve, in turn, through the ball-bearing, thrusts against the cone end and cone sleeve, thus closing it. In the other case the cam tends to leave the bowls; the belt, however, acting as a wedge, thrusts the cone halves as far apart as the cam will allow. To prevent the sleeve G from turning, flats are made on the top of the bowl pins which fit the slots in the casing. When one of the cones is closed and the other full open, there is an effective pulley ratio of 3 to 1; thus the extreme range of effective pulley diameter is 9 to 1.

The driving pulley makes 300 revs. per minute. The spindle makes from 25 to 219·5 with the single gear (whose ratio is 4 to 1) and from 4·96 to 43·1 with the treble gear (whose ratio is 20·376 to 1). This permits of a cutting speed of 30 feet per minute being obtained on all diameters between $\frac{1}{2}$ in. to $4\frac{1}{2}$ in. in the first case, and $2\frac{1}{2}$ in. to $23\frac{1}{2}$ in. in the second case.

It will be observed that the speeds overlap to some extent. This would be a disadvantage if it limited the range of speeds, but as $S_r = 40$ to 1, and it permits of work up to $4\frac{1}{2}$-in. diameter (which is the work a lathe of this size is usually engaged upon) being operated upon without having to engage the treble gear, and also allows a heavy cut to be taken at a high speed on hard material on work down to $2\frac{1}{2}$-in. diameter with the treble gear, there is possibly some advantage in the arrangement. By having the cone mounted on a side shaft which runs at a high rate of speed and is geared down to the spindle, it is possible for a machine with a belt drive of this type to take a comparatively heavy cut. The front spindle bearing is 4-in. diameter by 5 in. long. The back bearing is 3-in. diameter by $3\frac{5}{8}$ in. long. The end thrust of the spindle is taken on a ball-thrust bearing.

PARTICULARS OF WHEELS IN FAST HEADSTOCK.

Wheel.	Teeth.	Pitch.	Pitch diameter.	Centres.	Material.
			in.	in.	
C	35	6 d.p.	$5\frac{27}{32}$	$9\frac{1}{2}$	Cast iron
D	79	6 ,,	$13\frac{5}{32}$		
E	35	6 ,,	$5\frac{27}{32}$	$9\frac{1}{2}$	,,
F	79	6 ,,	$13\frac{5}{32}$		
A	19	5 ,,	$3\frac{51}{64}$	$9\frac{1}{2}$	,,
B	76	5 ,,	$15\frac{13}{64}$		
H	38	8 ,,	$4\frac{3}{4}$	—	,,
Worm wheels	43	$\frac{3}{8}$ c.p.	$5\frac{1}{8}$	$3\frac{13}{16}$	,,
Worms	—	$\frac{3}{8}$,,	$2\frac{1}{2}$	$2\frac{5}{8}$	
Indicator worm wheel	23	3 ,,	$2\frac{3}{4}$		,,

The saddle bears on the three top surfaces of the bed, the backward thrust being taken by the inside vertical face of the front surface, and a tapered slip at the outer face takes up the wear. The saddle is quite clear of the back vertical face of the bed, and is kept from lifting by the two parallel strips which fit on the bottom faces of the outer surfaces.

The annexed table gives some further particulars of these lathes.

Particulars of Langs' Variable Speed, Sliding, Surfacing, and Screw-cutting Lathes.

Height of centres.	Diameter of spindle, front bearing.	Ratio of gear.		Driving pulley.			Diameter admitted over saddle.	Width of bed over		Feeds per revolution of spindle.
		Single.	Treble.	Diameter.	Width.	Revs. per minute.		Top face.	Body.	
in.	in.			in.	in.		in.	in.	in.	in.
6	3	4	17	10	2½	300	8½	11¼	10¼	1/12, 1/24, 1/36, 1/48
8	4	4	20·37	14	3	,,	11½	15	13⅞	1/12, 1/24, 1/36, 1/48
10	5	4	27	16	3¾	,,	14¾	19	17⅝	1/8, 1/16, 1/24, 1/32
12	6	4	30	20	5	,,	18	22¼	20⅝	1/8, 1/16, 1/24, 1/32
15	7½	4	33	24	6	,,	22½	28	25⅝	1/8, 1/12, 1/16, 1/24

Lathe by the Lodge and Shipley Machine Tool Company.

The reversing gear, the feed-change gear, the saddle and apron for a 10-in. centre lathe by this firm have already been shown by Figs. 134 (p. 230), 147 (p. 241), and 157 (p. 258).

We now give, in Fig. 254 at the end of book, a front view of the all-gear headstock, and a general view of this machine (see Fig. 183), of which the following is a description:—

The design shown is that adopted by this firm after much experimenting. In their investigation they arrived at the conclusion that they could not take the desired cut at the proper speed, nor obtain a finishing or fine cut when driving direct, if the cone was mounted on the spindle in the ordinary way. They also found that the pull of the strap affected the alignment of the spindle. They accordingly adopted the following principles for guidance in this design: (*a*) The spindle was not to be subject to any change of alignment due to the pull of the strap; (*b*) the greater power was to be secured by the belt rather than by using high-gear ratios; (*c*) the finishing cuts were to be taken by the belt drive alone; (*d*) the lubrication of the bearings was to be automatic and positive.

The design of the headstock will be understood from Fig. 254, which shows the arrangement of the gears, with covers removed. The lathe is motor-driven, the motor being placed at the rear of the headstock and the armature shaft coupled directly to the shaft carrying the pinions J and G. These are keyed to, but free to slide together upon, the shaft. A sliding rod, having a rack formed upon it at one end, and provided with claws at the other, is employed in conjunction with a pinion and hand wheel on the vertical shaft to slide the pinions J and G, so that they may mesh respectively with the wheels K and H on the second-motion shaft. With this arrangement six spindle speeds are obtained from every motor speed, viz. first, G, H, I; second, J, K, H, I; third, G, H, I, A, B, E, F; fourth, J, K, H, I, A, B, E, F; fifth, G, H, I, C, D, E, F; sixth, J, K, H, I, C, D, E, F.

The belt-driven machine is identical with the above, except that a pulley takes the place of the wheel I on the sleeve, and the wheels G, H, J, and K are dispensed with. The drives are then : first, direct, the sleeve and F being clutched together; 2nd, through the double gear A, B, E, F; 3rd, through the double gear C, D, E, F. The driving pulley, together with the pinions A and C, are keyed to the sleeve, which is concentric with the spindle, but at no place touches it (being clear by about $\frac{1}{8}$th of an inch). The sleeve has its bearings formed in the two inner cheeks, whilst those of the spindle are in the outer cheeks of the headstock. The spindle is thus relieved of the pull of the belt. A clutch is formed on the projecting end of the sleeve nearest the front of the headstock which engages the clutch keyed to the spindle when driving direct. Clearance is allowed on the clutch so that any inequality of wear may not disturb the spindle alignment. Brass rings with buckets on their periphery are fixed to the spindle and sleeve at the centres of their journals. These buckets dip into deep oil wells in the headstock cheeks, and, when rotating, throw the oil over the journal. Suitable gutters are made in the bearings for the distribution of the oil, which, after passing through the bearings, returns to the well. Gauge glasses are fitted on the front of the cheeks to show the depth of oil in the wells. A similar system of lubrication is provided for the double-gear shaft. This shaft is hollow, and has a journal at each end, which runs in excentric bushes fitted in the headstock cheeks. A shaft passes through this hollow shaft to couple the excentric bushes, so that they are both turned simultaneously by the handle at the tail end of the headstock. The end thrust on the spindle is taken by the tail cheek and a large cast-iron collar keyed fast to the spindle. Between these are interposed two bronze washers placed on each side of a hardened steel washer of the same diameter.

Referring to Fig. 157 (p. 258), it will be observed that the bed has a flat surface within the front inverted vee to provide a support to the saddle and reduce the span of same. On the same figure is shown a wiper and oiler, which is fitted at each end of the carriage to prevent grit and cuttings getting between it and the bed.

The principal dimensions of these machines are given in the table on p. 395.

From 13½ to 24-in. centres two treble gears are furnished by the introduction of a pinion on the back-gear shaft, which gears with a wheel on a third-motion shaft. On this shaft is also keyed a pinion which meshes with an internal face-plate-wheel. The arrangement is identical with that commonly employed in treble-geared lathes, and the disengaging device is similar to that described above for the double-gear shaft.

LATHES BY HULSE, LTD.

We have shown the fast headstock, the saddle, the quick-traversing mechanism for same, the front and back rests, and the loose headstock for a 32-in. lathe by Figs. 251 (at end of book), 171–173 (pp. 279–283), Figs. 180 and 181 (pp. 296 and 298), and Fig. 198 (p. 309), *ante*.

Fig. 255 shows the gear bracket of the driving headstock of a powerful 36-in. centre duplex lathe recently (1906) completed by the above firm (Hulse & Co.). The machine was designed for sliding, surfacing, screw-cutting and tapering work. It had a bed 106 ft. 6 in. long, and was fitted with two driving and two sliding headstocks. With the headstocks in their normal position, 40 ft. could be admitted between the centres of each pair, and by sliding one pair (driving and sliding) of headstocks to the end of the bed, 80 ft. could be taken between the centres of the other pair. The driving headstock at the left-hand end of the bed was fixed and fitted with a 100-H.P. motor, whilst the other was moveable and provided with a motor of 80 H.P. Both motors were of the variable speed type, having a speed ratio of 2 to 1 and a maximum speed of 500 revs. per minute. The headstocks, however, were in other respects alike. The further reduction of speed to spindle was obtained through two treble gears having a ratio of 8·5 and 17 to 1, and four quintuple gears having a ratio of 34·87, 69·74, 133·87, and 267·75 to 1. These various ratios could be quickly secured by means of claw-racks and pinions (as shown in Fig. 255). The variation of speed at the motor, together with that obtained by the change-gears, gave a variation of speeds to the spindle of from 60 to 1 revs. per minute. All the gear-wheels were of mild steel and machine cut, and, with the exception of the faceplate wheel and pinion, all ran between bearings. The main spindle was of a very hard steel, and ran in hard gun-metal steps. The front journal was 12 in. diameter by $16\frac{1}{2}$ in. long. The rear journal was provided with collars, similar to those fitted upon a marine propellor thrust-block, to take the end thrust of the spindle. Four carriages were mounted on the bed, each fitted with duplex compound slide-rests which could be actuated transversely, independently, or simultaneously from the

LEADING DIMENSIONS OF THE LODGE AND SHIPLEY LATHES.

Equivalent height of centres	7 in.	8 in.	10 in.	12 in.	$13\frac{1}{2}$ in.	15 in.	18 to 24 in.
	in.	in.	in.	in.	in.	in.	in.
Diameter of pulley .	8	10	12	$15\frac{5}{16}$	17	18	24
Width of pulley . .	$3\frac{1}{2}$	4	5	$6\frac{1}{2}$	7	8	$9\frac{1}{2}$
Diameter of front bearing for spindle	$2\frac{5}{8}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$4\frac{3}{8}$	$4\frac{7}{8}$	$5\frac{3}{16}$	$6\frac{1}{4}$
Length of front bearing for spindle . .	$4\frac{3}{8}$	$5\frac{1}{8}$	$6\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{7}{8}$	$9\frac{1}{4}$	11
Diameter of back bearing for spindle	$1\frac{13}{16}$	$2\frac{1}{16}$	$2\frac{1}{2}$	$3\frac{7}{16}$	$3\frac{7}{8}$	$4\frac{1}{8}$	$5\frac{1}{4}$
Length of back bearing for spindle . .	$2\frac{7}{8}$	4	$4\frac{3}{8}$	$5\frac{3}{4}$	$5\frac{5}{8}$	$5\frac{3}{4}$	$8\frac{3}{4}$
Diameter of hole through spindle .	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{9}{16}$	$2\frac{1}{8}$
Diameter of pulley sleeve bearings . .	$2\frac{5}{8}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$4\frac{3}{8}$	$4\frac{7}{8}$	$5\frac{3}{16}$	$6\frac{1}{4}$
Length of pulley sleeve bearings . .	$2\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{1}{4}$	$5\frac{3}{4}$	$5\frac{5}{8}$	$5\frac{3}{4}$	7
Diameter of loose headstock spindle .	$1\frac{7}{16}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{3}{16}$	$3\frac{3}{16}$	5
Maximum spindle revs. per minute .	395	409	372	414	395	300	250
Minimum spindle revs. per minute .	20	20	17	13	12	8	2
Countershaft revs. per minute . . .	205 and 250	205 and 250	205 and 250	205 and 250	205 and 250	200 and 250	200 and 250
Double gear ratios .	3·04 and 9·6 to 1	3·12 and 9·3 to 1	3·12 and 9·8 to 1	3·69 and 13 to 1	3·75 and 13·8 to 1	3·9 and 14·4 to 1	2·94 and 8·29 to 1
Treble gear ratios . .	— —	— —	— —	— —	— —	22·4 and 62·3 to 1	21·97 and 61·9 to 1

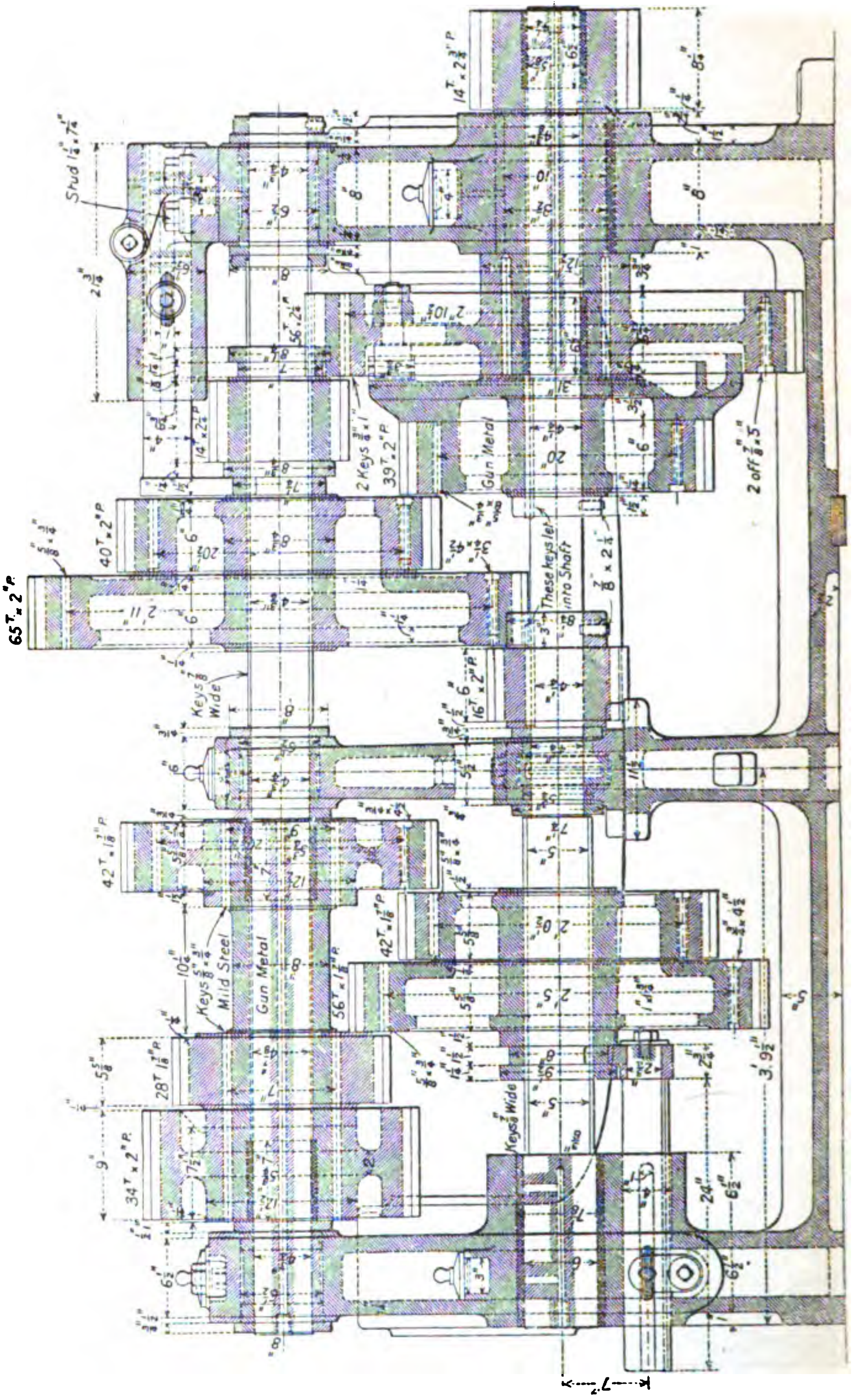


Fig. 255.—Arrangement of Gear for 86-in. Centre High-speed Lathe (Hulse).

front of the lathe. The rests (back and front) on the carriages nearest to the driving headstocks were provided with extra long swivel slides for boring purposes, etc. The front rests on the carriages nearest to the loose headstock were short, and the back rests on the same were of extra length, and fitted with an independent self-acting sliding motion by ratchet gear for turning the periphery of collars on propeller shafts, long tapers, etc. All the tool or top slides (8 in all) were of cast-steel. The screw-cutting and self-acting sliding motions were obtained through shafts in the centre of the bed, driven from the spindle by machine-cut steel change wheels at the end of the headstock, and other suitable gearing, to two simultaneously rotating nuts on the under side of each carriage. These nuts embraced two non-rotating screws, one on each side of the bed. Each carriage was also provided with a hand-traversing motion. The surfacing motion was effected by a swing frame and change wheels at the back of the saddles and the tapering motion by interchangeable worm-wheels. A quick-power traverse was also furnished to the saddles by a 20-H.P. motor at the left-hand end of the bed, and a 15 H.P. at the right-hand end. The change wheels at the end of the headstocks were disengaged by means of a lever and clutch, which when thrown over engaged the central shafts with the motor gears. The loose headstocks were furnished with hollow steel spindles, 7 in. diameter, and actuated by a screw and hand wheel for quick adjustment, or by a worm and wheel for forcing the centre into the work. These headstocks could be coupled to the nearest saddle and moved along the bed by hand or power. They were held in position by six bolts and clamping plates, in addition to a pawl on the head which engaged a stepped rack in the bed. The saddle for this lathe was illustrated by Fig. 166, and described on pp. 273–276, *ante*.

A few years ago a lathe of 60-in. centres was considered exceptional, but since the turbine has taken such an important place as a prime mover, lathes of 72 to 108-in. centres are by no means uncommon. At present these machines are of lighter proportions than those which we have given; and their dimensions would more nearly correspond to those of the "standard" 48 and 60-in. lathes respectively. There is, however, a general tendency to increase the power of each successive machine. In the latest 84-in. centre lathe by the above firm the headstock was fitted with a 100 H.P. motor, having a speed variation of 3 to 1. The arrangement of the driving gear was similar to that shown in Fig. 251 (at end of book), whereby two changes of treble and two of quintuple gearing gave a speed ratio to the spindle of 60 to 1. The main spindle was made of specially hard steel, and ran in phosphor bronze steps. The front journal was 15-in. diameter. The machine was arranged to swing 12 ft. over the carriages, of which there were two, each fitted with duplex rests.

The bed had four ways, the outer pair of which acted as supports only to the saddles, whilst the inner pair guided the same. It was 12 ft. 6 in. over the surfaces and 24 in. deep.

Lathe by Ernst Schiess.

The arrangement of the driving gear in the fast headstock for the 60-in. (1500 mm.) centre lathe by Ernst Schiess is shown in Fig. 252; it was described on p. 388. A general view of this machine is shown in the half-tone figure (Fig. 168 at the end of the book), and the saddle mechanism was shown in Fig. 169, p. 277. The slide rests in this machine have self-acting, reversible independent feeds in the longitudinal and transverse directions by means of splined shafts at the front and back of the bed. A longitudinal quick-power, as well as a hand-traverse motion, is also provided. The upper slides of the rests are of steel and interchangeable with one another. They differ in design, some being fitted with long swivelling slides, and others being of narrow construction for the purpose of facing crank cheeks.

The weight of the lathe complete, including two steady stays and one follower rest, etc., is approximately 126 tons.

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